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DEVELOPMENT OF A GRAPHICAL GENERAL SOLUTION FOR THERMAL CIRCULATION IN U-TUBE HYDRAULIC CIRCUITS.

BY LIEUT. COMMANDER CARL R. MIDTLYNG, U.S.N., MEMBER.

ABSTRACT.

The following treatment of thermal circulation employs a method of attack which, it is noted, is quite similar to that outlined by Markson, Ravese, and Humphreys *. Use of the U-tube circuit, the principal assumptions, the flow head loss—available head equation, and the assumption of a uniformly heated tube, are judged to be familiar devices in treating this subject. One innovation here present, however, is that all variables which effect circulation are introduced immediately and carried forward into a

* "A Method of Estimating the Circulation in Steam-Boiler-Furnace Circuits" by A. A. Markson, T. Ravese, and C. G. R. Humphreys—A.S.M.E. Tranactions of May, 1942

basic equation which lends itself to a direct ready solution. This work carries the investigation forward to the ultimate end, for, by repeated solutions in which the values of the various pertinent variables are changed in a systematic controlled manner, it is shown that there exists, a single set of characteristic curves which furnish full direct information for any U-tube circuit. This set of curves may be entered by calculating a key value, an operation which involves evaluaton of the hydraulic circuit flow resistances and key dimensions—a task having the magnitude of a preliminary step in other methods. This key value, in itself, has such direct significance, that its utility tends to eliminate the necessity of referring to the newly developed basic circulation curves. The U-tube circuit considered hereafter is called a "two-tube" circuit, and is given specific details of construction to facilitate calculations based on a certain class of marine boiler tube banks of special interest to the author. The net result, however, assumes a completely general character, applicable to any version of a U-tube hydraulic circuit. Extreme accuracy of calculation and plotting are not claimed in this work.

Development of Basic Equation.

The primary objective of the following is to investigate from purely theoretical considerations, the character of thermal circulation over an extended range of pressures and heat input rates. For this purpose, a basic equation relating heat input rate and mass rate of flow is sought by means of which the circulation characteristics of an ideal two-tube circuit of the type indicated by Figure 1 may be determined. A secondary objective lies in the application of this equation to boiler tube banks where the requisite degree of similarity with the two-tube circuit exists. For this purpose, the conditions necessary to justify the assumption of similarity are outlined. In order to apply an equation derived for the two-tube circuit to tube banks, an analysis is made of the basic general tube bank flow characteristics which are involved. Circulation characteristics are calculated substituting constants evaluated for a boiler tube bank into an equation derived for a two-tube circuit. The characteristic curves which are arrived at

must therefore be examined from two separate and distinct viewpoints—first, as indicative of the circulation characteristics of a two-tube circuit which has been proportioned to make applicable the constants used, and second, as characteristic curves of the tube bank in question. Proof of the validity of the latter or repeated demonstrations of the results so obtained is not undertaken. It is, however, concluded that the application is valid for certain types of tube banks, although the accuracy attained is somewhat questionable since the various flow loss coefficients cannot be accurately evaluated.

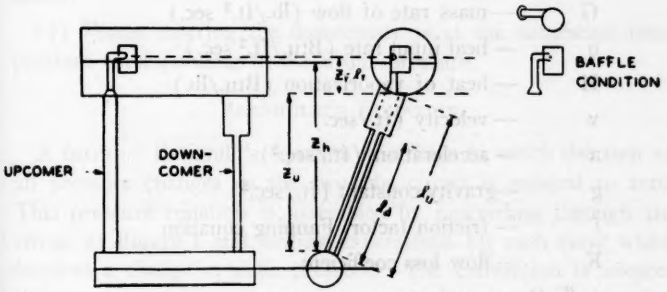


FIGURE 1—TWO-TUBE CIRCUIT.

The two-tube circuit indicated by Figure 1 is conceived to consist of an unheated downcomer, a uniformly heated upcomer, a steam drum, a water drum, and a baffle or steam separator device. A particular baffle arrangement is selected for treatment in order to develop the form in which conventional pressure loss coefficients will appear in a final equation. This arrangement contains all essential elements found in that class of high capacity marine boilers which are provided with ample external (unheated) downcomers, and lends itself readily to the setting up of an equation involving thermal flow or circulation within the circuit. Attention is confined to this one class of boilers since it is this class only which has sufficiently good circulation characteristics to warrant speculation as to ultimate limits in rating imposed by thermal circulation.

Symbols.

l_d — downcomer length (ft.)
l_u — upcomer length (ft.)
l_r — riser length (ft.)
Z_u — upcomer height (ft.)
Z_h — height, W.L. above downcomer exit (ft.)
Z_r — riser height (ft.)
D — internal tube diameter (ft.)
V — specific volume (ft.^{3}/lb.)
G — mass rate of flow (lb./ft.^{2} sec.)
q — heat input rate (Btu./ft.^{2} sec.)
H — heat of vaporization (Btu./lb.)
v — velocity (ft./sec.)
a — acceleration (ft./sec.^{2})
g —gravity constant (ft./sec.^{2})
f — friction factor, Fanning equation
K — flow loss coefficient
ϕ, Ψ, Θ — constants

SUBSCRIPTS.

d — downcomer
u — upcomer
r — riser
n — nozzle
f — fluid
b — baffle
fg — vaporization
m — mixture (steam and water)
A to R — constant designations
1 — entrance
2 — exit

ASSUMPTIONS.

(*a*) The upcomer is uniformly heated.

(*b*) Other parts of the circuit are perfectly insulated.

(*c*) Clockwise steady state flow exists.

(*d*) Water (only) enters the downcomer.

(*e*) Steam and water mixtures are stable—that is the tendency of steam bubbles to coalesce and rise relative to the surrounding water may be disregarded.

(*f*) A constant water level and steam drum pressure are maintained.

(*g*) Water entering the downcomer is at the saturation temperature corresponding to the drum pressure.

BERNOULLI'S EQUATION.

A form of Bernoulli's equation is written in which the sum of all pressure changes in the two-tube circuit is equated to zero. This pressure equation is assembled by proceeding through the circuit of Figure 1 and writing expressions for each event which involves a change in static pressure. The convention is adopted that terms involving an increase in static pressure when proceeding clockwise, are positive. For convenience in making subsequent transformations, the terms of this equation are arranged vertically in the left-hand column of Table I, which follows an explanation of the transformations presented below. The middle and right-hand columns of this table contain two successive transformations by which Bernoulli's equation is arranged to involve the two variables G_u and q in a useable form. These transformations are explained below.

FIRST TRANSFORMATION (SEE TABLE I).

The assumption that the upcomer is uniformly heated permits the specific volume of the fluid in the upcomer to be expressed as a linear function of the distance "l" from the upcomer entrance. The variables G_u and q are by definition not functions of l and therefore may enter such an expression and yet be treated as constants where an integration with respect to l is performed.

$$v = v_f + \left(\frac{\pi D_u \ell q}{i}\right)\left(\frac{4}{\pi D_u^2 G_u}\right)\left(\frac{1}{H_{fg}}\right)\left(\frac{v_{fg}}{i}\right) = v_f + \left(\frac{4 v_{fg} q}{G_u D_u H_{fg}}\right)\ell \qquad (1)$$

where

$(\pi D_u \ell q)$ = total heat input per second to point ℓ .

$(\pi D_u^2 G_u/4)$ = weight rate of flow at any point ℓ .

so that

$\left(\pi D_u \ell q\right)\left(\frac{4}{\pi D_u^2 G_u}\right)$ = heat input per pound of water at any point ℓ.

and

$\dfrac{(\pi D_u \ell q)}{(\pi D_u^2 G_u/4) H_{fg}}$ = lbs. of steam per pound of steam-water mixture at point ℓ .

The velocity " v " can be expressed as the product of the mass rate of flow G and the specific volume v

$$v\left(\frac{ft}{sec}\right) = G v \left(\frac{lb}{ft^2\ sec.}\right)\left(\frac{ft^3}{lb}\right) \qquad (2)$$

The differential $(a\,d\ell/g v)$ contains terms " a " and " v " which are linear functions of ℓ . Substitutions (1) and (2) above are employed.

$$a = \frac{dv}{dt}\ ,\quad d\ell = v\,dt,\quad a\,d\ell = \left(\frac{dv}{dt}\right)\left(v\,dt\right) = v\,dv$$

$$v = Gv = G\left(v_f + y\ell\right),\quad dv = G y\, d\ell$$

where

$$y = \left(\frac{4 v_{fg} q}{G_u D_u H_{fg}}\right)$$

then

$$\frac{a\,d\ell}{gv} = \frac{v\,dv}{gv} = \frac{G_u\left(v_f + y\ell\right)\left(G_u y\, d\ell\right)}{g\left(v_f + y\ell\right)} = \left(\frac{4 G_u v_{fg} q}{g D_u H_{fg}}\right) d\ell \qquad (3)$$

Employing substitutions (1), (2), and (3) above, terms in the left-hand column below are transformed to the forms found in the center column. G_u and q are selected as the dependent and independent variables, respectively, and are accordingly segregated. Other parts of each term of the middle equation are bracketed to form constants ϕ_a, ϕ_b,.....ϕ_R for convenience in handling.

SECOND TRANSFORMATION (SEE TABLE I).

The terms of the middle equation comprise an equation in G_u and q, but these two variables are not separable. Accordingly recourse is had to the substitution:

$$q/G_u = X \tag{4}$$

yielding the equation of the right-hand column. The variables G_u and X are now separable to the extent that G_u is expressed as a function of X, the equation reducing to the form—

$$G_u = \sqrt{\frac{\frac{z_n}{v_f} - \frac{\phi_H}{X} \ell n\,(1+\phi_1 X) - \frac{z_r}{v_f + \phi_R X}}{\psi_A + \psi_B X}} \tag{5}$$

where

$$\phi_H = \left(\frac{z_u}{4}\right)\left(\frac{D_u}{\ell_u}\right)\left(\frac{H_{fg}}{v_{fg}}\right) \tag{6}$$

$$\phi_1 = 4\left(\frac{\ell_u}{D_u}\right)\left(\frac{v_{fg}}{H_{fg}\, v_f}\right) \tag{7}$$

$$\phi_R = 4\left(\frac{\ell_u}{D_u}\right)\left(\frac{v_{fg}}{H_{fg}}\right) \tag{8}$$

$$\psi_A = \left(\phi_A + \phi_B - \phi_C + \phi_D + \phi_E + \phi_J + \phi_L + \phi_N + \phi_P\right)$$

$$\psi_A = \frac{v_{fg}}{2g}\left[\left(d K_1 + d K_2 + \frac{4 f_d\, \ell_d}{D_d}\right)\left(\frac{D_u}{D_d}\right)^4 + \left(1 + {}_u K_1 + \frac{4 f_u\, \ell_u}{D_u}\right) + \left(1 + {}_r K_1 + \frac{4 f_r\, \ell_r}{D_r}\right)\left(\frac{D_u}{D_r}\right)^4 + \left(1 + {}_n K_1 + {}_n K_2\right)\left(\frac{D_u}{D_n}\right)^4\right] \tag{9}$$

$$\psi_B = \left(\phi_F + \phi_G + \phi_K + \phi_m + \phi_o + \phi_Q\right)$$

$$\psi_B = \frac{v_{fg}}{g H_{fg}}\left[4 f_u \left(\frac{\ell_u}{D_u}\right)^2 + 4\left(\frac{\ell_u}{D_u}\right) + 2\left(1 + {}_r K_1 + \frac{4 f_r\, \ell_r}{D_r}\right)\left(\frac{\ell_u}{D_u}\right)\left(\frac{D_u}{D_u}\right)^4 + 2\left(1 + {}_n K_1 + {}_n K_2\right)\left(\frac{\ell_u}{D_u}\right)\left(\frac{D_u}{D_n}\right)^4\right] \tag{10}$$

TABLE I.

1. Gravity effect in downcomer leg.		
$+ z_h / v_f$	$+ z_h / v_f$	$+ z_h / v_f$
2. Downcomer entrance.		
$-\frac{(1+{}_dK_1)\, v_d^2}{2g\, v_f}$	$-\left[\frac{(1+{}_dK_1)\, D_U^4\, v_f}{2g\, D_d^4}\right] G_U^2$	$-\phi_A G_U^2$
3. Downcomer friction.		
$-\frac{2 f_d\, v_d^2\, l_d}{g D_d\, v_f}$	$-\left[\frac{2 f_d\, D_U^5\, v_f\, l_d}{g\, D_d^5}\right] G_U^2$	$-\phi_B G_U^2$
4. Downcomer exit.		
$+\frac{(1-{}_dK_2)\, v_d^2}{2g\, v_f}$	$+\left[\frac{(1-{}_dK_2) D_U^4\, v_f}{2g\, D_d^4}\right] G_U^2$	$+\phi_C G_U^2$
5. Upcomer entrance.		
$-\frac{(1+{}_UK_1)\, v_U^2}{2g\, v_f}$	$-\left[\frac{(1+{}_UK_1) v_f}{2g}\right] G_U^2$	$-\phi_D G_U^2$
6. Upcomer friction.		
$-\int_0^{l_U} \frac{2 f_U\, v_U^2\, d\ell}{g\, D_U\, v_U}$	$-\left(\frac{2 f_U\, v_f\, l_U}{g\, D_U}\right) G_U^2 - \left(\frac{4 f_U\, v_{fg}\, l_U^2}{g\, D_U^2\, H_{fg}}\right) G_U q$	$-\phi_E G_U^2 - \phi_F G_U^2 x$
7. Acceleration in upcomer.		
$-\int_0^{l_U} \frac{a\, d\ell}{g\, v_U}$	$-\left(\frac{4\, v_{fg}\, l_U}{g\, D_U\, H_{fg}}\right) G_U q$	$-\phi_G G_U^2 x$
8. Gravity effect in upcomer.		
$-\frac{z_U}{l_U}\int_0^{l_U} \frac{d\ell}{v_U}$	$-\frac{\left(\frac{z_U\, D_U\, H_{fg}}{4 l_U\, v_{fg}}\right)}{q/G_U} \ln\left[1+\left(\frac{4 v_{fg}\, l_U}{D_U H_{fg} v_f}\right)\frac{q}{G_U}\right]$	$\frac{\phi_H}{x} \ln(1+\phi_I x)$
9. Baffle riser entrance.		
$-\frac{(1+{}_rK_1)\, v_r^2}{2g\, v_m}$	$-\left(\frac{(1+{}_rK_1) D_U^4\, v_f}{2g\, D_r^4}\right) G_U^2 - \left(\frac{2(1+{}_rK_1) D_U^3\, v_{fg}\, l_U}{g\, D_r^4\, H_{fg}}\right) G_U q$	$-\phi_J G_U^2 - \phi_K G_U^2 x$
10. Baffle riser friction.		
$-\int_0^{l_r} \frac{2 f_r\, v_r^2\, d\ell}{g\, D_r\, v_m}$	$-\left(\frac{2 f_r\, D_U^3\, v_f\, l_U}{g\, D_r^5}\right) G_U^2 - \left(\frac{8 f_r\, v_{fg}\, l_r\, l_U D_U^3}{g\, D_r^5\, H_{fg}}\right) G_U q$	$-\phi_L G_U^2 - \phi_M G_U^2 x$
11. Separator nozzle entrance.		
$-\frac{(1+{}_nK_1)\, v_n^2}{2g\, v_m}$	$-\left[\frac{(1+{}_nK_1) D_U^4\, v_f}{2g\, D_n^4}\right] G_U^2 - \left[\frac{2(1+{}_nK_1) D_U^3\, v_{fg}\, l_U}{g\, D_n^4\, H_{fg}}\right] G_U q$	$-\phi_N G_U^2 - \phi_O G_U^2 x$
12. Separator nozzle exit.		
$-\frac{{}_nK_2\, v_n^2}{2g v_m}$	$-\left(\frac{{}_nK_2 D_U^4 v_f}{2g\, D_n^4}\right) G_U^2 - \left(\frac{2\, {}_nK_2\, D_U^3 v_{fg}\, l_U}{g D_n^4 H_{fg}}\right) G_U q$	$-\phi_P G_U^2 - \phi_Q G_U^2 x$
13. Gravity in baffle riser.		
$-z_r / v_f$	$-\frac{z_r}{v_f + \left(\frac{4 v_{fg}\, l_U}{D_U H_{fg}}\right)\frac{q}{G_U}}$	$\frac{-z_r}{v_f + \phi_R x}$

Equation (5) constitutes a perfectly useable equation in G_u and X by use of which the thermal circulation performance of a two-tube circuit may be calculated. An equation expressing G_u as a function of q would be the most desirable form, but is not obtainable by the method used. The ratio q/G_u or X when multiplied by another ratio (a constant) has more direct significance than the quantity G_u. Let R be called the "circulation factor", and defined as—

$$R = \frac{\text{(weight rate of steam from upcomer)}}{\text{(weight rate of water to upcomer)}} \qquad (11a)$$

or, 100 R = per cent steam (by weight) in the steam water mixture issuing from the upcomer.

An index of circulation finding common use is known as the "circulation ratio", and is defined as:

$$R_c = \frac{\text{(weight rate of water flow to upcomers)}}{\text{(weight rate of steam flow from upcomers)}} \qquad (11b)$$

R_c is the reciprocal of R, giving the relation

$$R = \frac{1}{R_c} \qquad (11c)$$

In this work, R is not again involved, a fact which must be borne in mind by those accustomed to its use.

now

$(\pi D_U l_U q)$ = rate of total heat input to upcomer.

$\left(\frac{G_U^2 \pi D_U^2}{4}\right)$ = weight rate of flow from upcomer.

so that

$\left(\frac{\pi D_U l_U q}{\pi D_U^2 G_U/4}\right)$ = heat input per lb. of mixture issuing from the upcomer.

Dividing the latter quantity by H_{fg} gives the pounds of steam per pound of steam-water mixture issuing from the upcomer, or—

$$R = \left(\frac{\pi D_U l_U q}{}\right)\left(\frac{4}{\pi D_U^2 G_U}\right)\left(\frac{1}{H_{fg}}\right) = \left(\frac{4 l_U}{D_U H_{fg}}\right)\left(\frac{q}{G_U}\right) = \left(\frac{4 l_U}{D_U H_{fg}}\right) x \qquad (12)$$

Substituting the relation (12) in (5) yields:

$$q = \left(\frac{D_U H_{fg}}{4 l_U}\right) R \sqrt{\frac{Z_h/v_f - \frac{Z_U}{v_{fg} R} \ln\left(1 + \frac{v_{fg} R}{v_f}\right) - \frac{Z_r}{v_f\left(1 + \frac{v_{fg} R}{v_f}\right)}}{\theta_A + \theta_B R}} \qquad (13)$$

where

$$\theta_A = \theta_A \qquad (14)$$

and

$$\theta_B = \left(\frac{D_U}{l_U}\right)\left(\frac{v_{fg}}{4 f}\right)\left[\begin{matrix}\text{BRACKETED} \\ \text{TERMS OF (10)}\end{matrix}\right] \qquad (15)$$

Although the reverse would be more desirable, equation (13) defines q as a function of R, and may be considered an alternate form of the basic equation (5).

Application of Basic Equation to Tube Banks.

In applying the two-tube equation to a tube bank, there are two conditions which must exist. They are:

(*a*) All heated generating tubes function as upcomers at all heat input rates.

(*b*) The relation between average upcomer mass rate of flow and average upcomer heat input rate is given by equation (5). If the tube bank is provided with adequate external downcomers, the first condition will be met at high heat input rates, although, at low heat input rates, this is not true unless a solid plate "cyclone" type of baffling apron, such as indicated in Figures 1 and 2, is present. The validity of assuming the second condition to be met is not easy to prove or disprove, but, since accuracy is not claimed for application to tube banks, a positive assumption will be made without qualification. On the basis then that both conditions are met, consideration is given to applying equations (5) or (13) to a tube bank of the type indicated in Figure 2. The basic general variables are mass rate of flow G_u and heat transfer rate q. These variables are defined as:

$$G_U = \frac{\text{total weight rate of flow to upcomers}}{\text{total upcomer flow area}} \left(\frac{\text{lb}}{\text{ft}^2\ \text{sec.}}\right) \tag{16}$$

$$q = \frac{\text{total heat input, Btu/sec.}}{\text{total waterside heading surface}} \left(\frac{\text{Btu}}{\text{ft}^2\ \text{sec.}}\right) \tag{17}$$

Their relationship is established by the following general circuit characteristics which are:

(*a*) The total available head.

(*b*) The average upcomer length.

(*c*) The upcomer diameter.

(*d*) The average upcomer inclination.

(*e*) The effective riser height.

(*f*) The effective downcomer overall pressure loss coefficient.

(*g*) The effective baffle overall pressure loss coefficient.

Items (*a*) to (*g*), inclusive, furnish necessary information to calculate the circulation characteristics of the circuit and are perfectly general in that they might be applicable to any combination of numbers of upcomers and downcomers. To investigate means of evaluating items (*f*) and (*g*) above, resort is had to an analysis of equation (5). Inspection of terms of this equation indicates that circuit proportions and not dimensions are involved. The condensed expression ($\Psi_A + \Psi_B X$) of equation (5) is a multiplier which combined with G^2_u, gives a product which is the sum of all pressure changes in the circuit chargeable to flow. The manner in which standard pressure loss coefficients enter into this equation are more clearly seen by the following rearrangement of Ψ_A and Ψ_B:

$$\psi_A = \frac{v_i}{2g}\left[\kappa_d\left(\frac{D_U}{D_d}\right)^4 + (\kappa_U)_i + \kappa_r\left(\frac{D_U}{D_r}\right)^4 + \kappa_n\left(\frac{D_U}{D_n}\right)^4\right] \tag{18}$$

Where

$$K_d = \left(_dK_1 + {}_dK_2 + 4f_d \frac{\ell_d}{D_d}\right) \quad (19)$$

$$(K_U)_1 = \left(1 + {}_UK_1 + 4f_U \frac{\ell_U}{D_U}\right) \quad (20)$$

$$K_r = \left(1 + {}_rK_1 + 4f_r \frac{\ell_r}{D_r}\right) \quad (21)$$

$$K_n = \left(1 + {}_nK_1 + {}_nK_2\right) \quad (22)$$

$$\psi_B = \left(\frac{v_{fg}}{g H_{fg}}\right)\left(\frac{\ell_U}{D_U}\right)\left[(K_U)_2 + 2K_r\left(\frac{D_U}{D_r}\right)^4 + 2K_n\left(\frac{D_U}{D_n}\right)^4\right] \quad (23)$$

where

$$(K_U)_2 = \left(\frac{4f_U \ell_U}{D_U} + 4\right) \quad (24)$$

A rearranged form for $G_U^2\left(\psi_A + \psi_B X\right)$ is thus arrived at:

$$\Delta P = \frac{G_U^2 v_f}{2g}\left[K_d\left(\frac{D_U}{D_d}\right)^4 + (K_U)_1 + K_r\left(\frac{D_U}{D_r}\right)^4 + K_n\left(\frac{D_U}{D_n}\right)^4\right] + 2G_U^2\left(\frac{q}{G_U}\right)\left(\frac{v_{fg}}{g H_{fg}}\right)\left(\frac{\ell_U}{D_U}\right)\left[(K_U)_2 + K_r\left(\frac{D_U}{D_r}\right)^4 + K_n\left(\frac{D_U}{D_n}\right)^4\right] \quad (25)$$

In the expression above, the terms $\left[K_d\left(\frac{D_U}{D_d}\right)^4\right]$ AND $\left[K_r\left(\frac{D_U}{D_r}\right)^4 + K_n\left(\frac{D_U}{D_n}\right)^4\right]$

constitute what may be considered possible forms of overall pressure loss coefficients for the downcomer and baffle arrangement respectively. The general method of evaluating overall coefficients is indicated in the following example in which the large tube bank of the boiler indicated by Figure 2 below is selected for calculation of circulation characteristics through use of equation (5). The necessary degree of similarity with the two-tube circuit is deemed to exist.

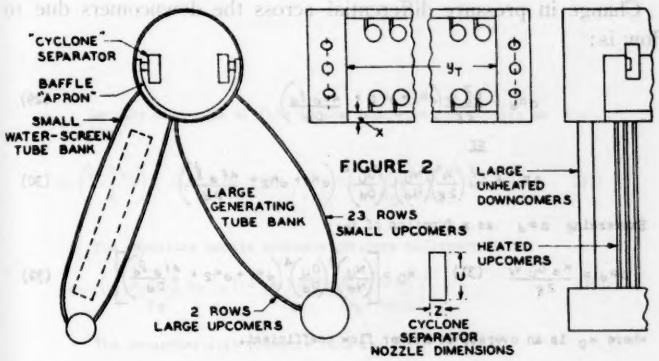


FIGURE 2

Number of Upcomers.

Where the tube bank contains two sizes of upcomer tubes—a comparatively small number of large size tubes in the tube rows adjacent to the furnace, and the remainder of a smaller uniform diameter—the tube bank is assumed to contain a corrected total number of small size tubes only. Let:

N_L = number of large tubes of diameter D_L (actual)

N_a = number of small tubes of diameter D_a (actual)

N_U = corrected number of small tubes of diameter D_U

then

$$N_U = \frac{N_L D_L^2 + N_a D_a^2}{D_U^2} \qquad (26)$$

Overall Downcomer Flow Coefficient

Let: N_d = number of downcomers of diameter D_d

N_U = number of upcomers of diameter D_U

then

$$G_U \left(\frac{\pi D_U^2}{4}\right) N_U = G_d \left(\frac{\pi D_d^2}{4}\right) N_d \qquad (27)$$

$$G_d = \left(\frac{N_U}{N_d}\right)\left(\frac{D_U}{D_d}\right)^2 G_U \qquad (28)$$

Change in pressure differential across the downcomers due to flow is:

$$\Delta P_d = \frac{G_d^2 v_f}{2g}\left(dK_1 + dK_2 + \frac{4f_d l_d}{D_d}\right) \qquad (29)$$

or

$$\Delta P_d = G_U^2\left(\frac{v_f}{2g}\right)\left(\frac{N_U}{N_d}\right)^2\left(\frac{D_U}{D_d}\right)^4\left(dK_1 + dK_2 + \frac{4f_d l_d}{D_d}\right) \qquad (30)$$

Expressing ΔP_d as a function of G_U –

$$\Delta P_d = \frac{K_d G_U^2 v_f}{2g} \quad (31) \qquad K_D = \left[\left(\frac{N_U}{N_d}\right)^2\left(\frac{D_U}{D_d}\right)^4\left(dK_1 + dK_2 + \frac{4f_d l_d}{D_d}\right)\right] \qquad (32)$$

where K_D is an overall downcomer flow coefficient.

Overall Baffle Flow Coefficient.

The flow pressure differential across the baffle arrangement in the tube bank being considered is assumed to consist of an entrance loss to the baffle "apron", a friction loss in the apron, an entrance loss to the separator nozzles, and a separator nozzle exit loss. Since the large tube bank of Figure 2 generates an average of 65 per cent of the total steam generated, the length "y_T" of the apron, and the number of separator cans "N_{ST}" are arbitrarily reduced to 65 per cent of their actual values (y and N_s) and the reduced baffling considered to function only with the large tube bank. Let the mass rate of flow in the baffle apron be G_R. Then—

$$G_r\left(2xy\right) = G_U\left(\frac{\pi D_U^2}{4}\right)N_U, \; G_r = G_U\left(\frac{\pi D_U^2 N_U}{8xy}\right) \qquad (33)$$

The baffle apron entrance pressure differential is:

$$\Delta P = \frac{\left(1 + rK_1\right)G_r^2 v_m}{2g} = \frac{\left(1 + rK_1\right)}{2g}\left(\frac{\pi D_U^2 N_U}{8xy}\right)^2 G_U^2 v_m \qquad (34)$$

The baffle apron friction pressure differential is:

$$\frac{f_r v_r^2 l_r}{2gM v_m} = \frac{f_r G_r^2 v_m l_r}{2g\left[\frac{xy}{2(x+y)}\right]} = \frac{f_r G_r^2 v_m l_r (x+y)}{8xy} \qquad (35)$$

$$= \left(\frac{f_r l_r (x+y)}{8xy}\right)\left(\frac{\pi D_U^2 N_U}{8xy}\right)^2 G_U^2 v_m \qquad M = \text{hydraulic radius}$$

Let the mass rate of flow in the separator can nozzles be G_n , then –

$$G_U \left(\frac{\pi D_U^2 N_U}{4}\right) = G_n N_S U Z \quad (36) , \qquad G_n = \left(\frac{N_U}{N_S}\right)\left(\frac{\pi D_U^2}{4 U Z}\right) G_U \quad (37)$$

The separator nozzle entrance pressure differential is:

$$\Delta P = \frac{(1+n K_1)\, G_n^2\, v_m}{2g} = \frac{(1+n K_1)\, G_U^2\, v_m}{2g}\left(\frac{N_U}{N_S}\right)^2\left(\frac{\pi D_U^2}{4UZ}\right)^2 \quad (38)$$

The separator exit pressure loss is:

$$\Delta P = \frac{n K_2\, G_U^2\, v_m}{2g} = \frac{n K_2\, G_U^2\, v_m}{2g}\left(\frac{N_U}{N_S}\right)^2\left(\frac{\pi D_U^2}{4UZ}\right)^2 \quad (39)$$

Assembling all baffle flow terms –

$$\Delta P_b = G_U^2\left(\frac{v_m}{2g}\right)\left(\frac{\pi D_U^2 N_U}{8 x y}\right)^2\left(1+r K_1 + \frac{f_r\, l_r\,(x+y)}{x y}\right) + \left(\frac{N_U}{N_S}\right)^2\left(\frac{\pi D_U^2}{4UZ}\right)^2\left(1+n K_1 + n K_2\right) \quad (40)$$

Expressing v_m as a function of G_U and q

$$v_m = v_f + \frac{4\, v_{fg}\, l_U\, q}{G_U\, D_U\, H_{fg}}$$

then

$$\Delta P_b = G_U^2\left(\frac{v_f + \frac{4 v_{fg}\, l_U\, q}{G_U D_U H_{fg}}}{2g}\right)\left[\left(\frac{\pi D_U^2 N_U}{8 x y}\right)^2\left(1+r K_1 + 2\,\frac{f_r\, l_r\,(x+y)}{x y}\right) + \left(\frac{N_U}{N_S}\right)^2\left(\frac{\pi D_U^2}{4UZ}\right)^2\left(1+n K_1 + n K_2\right)\right] \quad (41)$$

The overall baffle flow coefficient then becomes

$$K_B = \left[\left(\frac{\pi D_U^2 N_U}{8 x y}\right)^2\left(1+r K_1 + 2\,\frac{f_r\, l_r\,(x+y)}{x y}\right) + \left(\frac{N_U}{N_S}\right)^2\left(\frac{\pi D_U^2}{4UZ}\right)^2\left(1+n K_1 + n K_2\right)\right] \quad (42)$$

A revised form for $(\psi_A + \psi_B X)$ then becomes:

$$\Delta P_{fl} = G_U^2 \left(\frac{v_f}{2g}\right)\left[K_D + (K_U)_1 + K_B\right] + G_U^2 \left(\frac{2 v_{fg} l_U}{g D_U H_{fg}}\right)\left(\frac{q}{G_U}\right)\left[(K_U)_2 + K_B\right] \tag{43}$$

where

$$\frac{\left(\frac{4 v_{fg} l_r q}{G_U D_U H_{fg}}\right)}{2g} = 2\left(\frac{v_{fg}}{g H_{fg}}\right)\left(\frac{l_U}{D_U}\right)\left(\frac{q}{G_U}\right) \tag{44}$$

$$(K_U)_1 = 1 + \upsilon K_1 + \left(\frac{4 f_U l_U}{D_U}\right) \tag{45}$$

$$(K_U)_2 = \frac{1}{2}\left(\frac{4 f_U l_U}{D_U} + 4\right) \tag{46}$$

and K_D and K_B are given by (32) and (42), respectively.

Equation (5) is transformed into a form more suitable for direct application to boiler tube banks by methods indicated above.

$$G_U = \sqrt{\frac{\frac{z_h}{v_f} - \left(\frac{z_U}{v_f}\right)\frac{\ln(1+\phi_1 X)}{\phi_1 X} - \frac{z_r}{v_f(1+\phi_1 X)}}{\frac{v_f}{2g}\left[K_D + (K_U)_1 + K_B\right] + 2\left(\frac{v_{fg}}{g H_{fg}}\right)\left(\frac{l_U}{D_U}\right)\left[(K_U)_2 + K_B\right] X}} \tag{47}$$

where

$$\phi_1 = 4\left(\frac{l_U}{D_U}\right)\left(\frac{v_{fg}}{H_{fg} v_f}\right) \tag{7}$$

K's are as defined above.

Sample Solution for Tube Bank.

The general method of solving for tube bank circulation characteristics is indicated by the following example in which the large tube bank of Figure 2 is considered at an absolute pressure of 620 psi. In this solution, equation (47) is employed with a series of assumed values of X (q/G) covering the entire range R = 0 to R = 1.0. Results are plotted as characteristic curves in Figure 3.

Pertinent Data.

z_h = 11 ft.

z_U = 9.7 ft.

z_f = 1.8 ft.

l_U = 12 ft.

D_U = 1/12 ft.

D_L = 2/12 ft.

N_a = 1805

N_L = 56

N_U = 2029

l_d = 12 ft.

D_d = 3.814/12 ft.

N_d = 28

N_{ST} = 22

N_S = 14.3

y_T = 126.5/12 ft.

x = 3.2/12 ft.

y = 82.2/12 ft.

U = 2.5/12 ft.

z = 10.5/12 ft.

v_f = 0.0202 ft.^{3}/lb. } 620 psi abs.

v_{fg} = 0.7217 ft.^{3}/lb. } 620 psi abs.

H_{fg} = 725.1 Btu/lb. } 620 psi abs.

$_dK_1$ = 0.2

$_dK_2$ = 0.2

$_UK_1$ = 0.2

$_rK_1$ = 0.2

$_nK_1$ = 0.2

$_nK_2$ = 0.2

f_d = 0.006 } Fanning equation

f_u = 0.006 } Fanning equation

f_r = 0.006 } Fanning equation

es = % steam generated in large tube bank

TABLE II—EVALUATION OF CONSTANTS.

$Z_h/v_f = 11/0.0202 =$	545
$Z_U/v_f = 97/0.0202 =$	480
$Z_r/v_f = 18/0.0202 =$	89.1
$\phi_I = 4\left(\frac{l_U}{D_U}\right)\left(\frac{v_{fg}}{v_f H_{fg}}\right) = \frac{4 \times 144 \times 0.7217}{0.0202 \times 725.1} =$	28.45
$v_f/2g = \frac{0.0202}{2 \times 32.2} =$	3.14/104
$2\left(\frac{v_{fg}}{g H_{fg}}\right)\left(\frac{l_U}{D_U}\right) = 2\left(\frac{0.7217}{32.2 \times 725.1}\right)\left(\frac{12 \times 12}{1}\right) =$	0.00891
$N_U = \frac{N_L D_L^2 + N_{a_1} D_U^2}{D_U^2} = \frac{56 \times \left(\frac{2}{12}\right)^2 + 1805\left(\frac{1}{12}\right)^2}{\left(\frac{1}{12}\right)^2} =$	2029
$\left(\frac{N_U}{N_d}\right)^2\left(\frac{D_U}{D_d}\right)^4 = \left(\frac{2029}{28}\right)^2 \times \left(\frac{1}{3.814}\right)^4 = \left(\frac{3230}{1}\right)\left(\frac{1}{212}\right) =$	24.8
$\left(\frac{4 f_d l_d}{D_d}\right) = \frac{4 \times 6 \times 12 \times 12}{1000 \times 3.814} =$	0.907
$y = 0.65\, y_T = (0.65)\left(126\ 5/12\right) =$	82.2/12
$\left(\frac{\pi D_U^2 N_U}{8 X y}\right)^2 = \left(\frac{\pi \times 2029 \times 144}{144 \times 8 \times 82.2 \times 32}\right)^2 =$	9.18
$\frac{2 f_r l_r (x+y)}{X y} = \frac{2 \times 6 \times 18 (82.2 + 3.2) \times 144}{1000 \times 82.2 \times 3.2 \times 12} =$	0.084
$N_S = 0.65 \times N_{ST} = 0.65\,(22) =$	14.3
$\left(\frac{N_U}{N_S}\right)^2\left(\frac{\pi D_U^2}{4 U Z}\right)^2 = \left(\frac{2029}{14.3}\right)^2\left(\frac{\pi \times 144}{144 \times 4 \times 10.5 \times 2.5}\right)^2 =$	18.2
$\frac{4 f_U l_U}{D_U} = \frac{4 \times 6 \times 12 \times 12}{1000} =$	3.46

TABLE II—EVALUATION OF CONSTANTS.

Ψ_A

$$K_d = \left(\frac{N_U}{N_d}\right)^2 \left(\frac{D_U}{D_d}\right)^4 \left(dK_1 + dK_2 + \frac{4 f_d l_d}{D_d}\right) = 24.8\,(0.2 + 0.2 + 0.907) = 32.4$$

$$(K_U)_1 = \left(1 + UK_1 + 4\,\frac{f_U l_U}{D_U}\right) = (1 + 0.2 + 3.46) = 4.66$$

$$K_B = \left[\left(\frac{\pi D_U^2 N_U}{8 X y}\right)^2 \left(1 + rK_1 + \frac{2 f_r l_r (x+y)}{xy}\right) + \left(\frac{N_U}{N_S}\right)^2 \left(\frac{\pi D_U^2}{4 U Z}\right)^2 (1 + nK_1 + nK_2)\right] =$$

$$= \left[\underbrace{(9.18)\,(1 + 0.2 + 0.084)}_{11.8} + \underbrace{(18.2)\,(1 + 0.2 + 0.2)}_{25.5}\right] = 37.3$$

$$\Psi_A = \left(\frac{V_f}{2g}\right)\left[K_D + (K_U)_1 + K_B\right] = \frac{3.14}{10^4}\left(\frac{32.4 + 4.66 + 37.3}{74.4}\right) = 0.0234$$

Ψ_B

$$(K_U)_2 = \frac{1}{2}\left(\frac{4 f_U l_U}{D_U} + 4\right) = \frac{1}{2}\,(3.46 + 4) = 3.73$$

$$\Psi_B = 2\left(\frac{V_{fg}}{g H_{fg}}\right)\left(\frac{l_U}{D_U}\right)\left[(K_U)_2 + K_B\right] = (0.00891)\,(3.73 + 37.3) = 0.365$$

TABLE II—CALCULATIONS.

	x	0.005	0.010	0.015	0.025	0.040	0.060	0.090
1	$\phi_I x$	0.142	0.285	0.427	0.711	1.14	1.71	2.56
2	$1+\phi_I x$	1.142	1.285	1.427	1.711	2.14	2.71	3.56
3	$\ln(1+\phi_I x)$	0.133	0.251	0.356	0.536	0.760	1.005	1.27
4	$\ln(1+\phi_I x)/\phi_I x$	0.937	0.881	0.833	0.754	0.667	0.588	0.497
5	$\frac{z_U}{v_f}$ ④	449	422	399	362	318	282	239
6	$\frac{z_r}{v_f}$ / ②	78	69.3	62.6	52.1	41.7	32.9	25.0
7	⑤ + ⑥	527	491.3	461.6	414.1	359.7	314.9	264.0
8	$\frac{z_h}{v_f}$ − ⑦	18	53.7	83.4	130.9	185.3	230.1	281.0
9	$\psi_B x$	.00182	.00365	.00548	.00913	.0146	.0219	.0328
10	$\psi_A + \psi_B x$	.0252	.0271	.0289	.0325	.0380	.0453	.0562
11	⑧/⑩	715	1980	2885	4020	4880	5080	5000
12	G = √⑪	26.8	44.6	53.8	63.5	70.0	71.3	70.8
13	q = Gx	0.134	0.446	0.808	1.59	2.80	4.27	6.37
14	$R = x\left(4\ell_U/D_U H_{fg}\right)$	.00398	.00795	.0119	.0199	.0318	.0476	.0716

$\phi_I = 28.45$, $z_U/v_f = 480$, $z_r/v_f = 89.1$, $z_h/v_f = 545$

$\psi_B = 0.385$, $\psi_A = 0.0234$, $\frac{4\ell_U}{D_U H_{fg}} = \left(\frac{4 \times 12 \times 12}{725.1}\right) = 0.795$

$(x)_{MAX.} = \frac{D_U H_{fg}}{4\ell_U} \times 1.0 = 1.26$

TABLE II.

	x	0.120	0.15	0.27	0.4	0.8	1.0	1.26
1	ϕ_I x	3.41	4.26	7.68	11.35	22.7	28.45	35.8
2	$1+\phi_I$ x	4.41	5.26	8.68	12.35	23.7	29.45	36.8
3	$f_n(1+\phi_I x)$	1.45	1.66	2.16	2.51	3.17	3.38	3.60
4	③/ϕ_I x	0.426	0.390	0.282	0.221	0.1395	0.119	0.1003
5	$\frac{z_U}{v_f}$ x ④	205	187	137	106	67.0	57.0	48.3
6	$\frac{z_r}{v_f}$ /②	20.2	16.95	10.3	7.21	3.76	3.02	2.42
7	⑤ + ⑥	225.2	204	147.3	113.2	70.6	60.0	50.7
8	$\frac{z_h}{v_f}$ - ⑦	319.8	341.0	397.7	431.8	474.2	485.0	494.3
9	Ψ_B x	.0438	.0548	0.0985	.0146	.0292	0.365	0.460
10	$\Psi_A + \Psi_B$ x	.0672	.0782	0.1219	0.169	0.315	0.388	0.483
11	⑧/⑩	4770	4370	3260	2550	1505	1250	1022
12	G = $\sqrt{⑪}$	69.2	66.2	57.2	50.6	38.8	35.4	32.0
13	q = G x	8.30	9.93	15.4	20.2	31.1	35.4	40.3
14	$R = x\left(\frac{4 f_U}{D_U H_{fg}}\right)$	0.0955	0.119	0.214	0.318	0.635	0.795	1.00

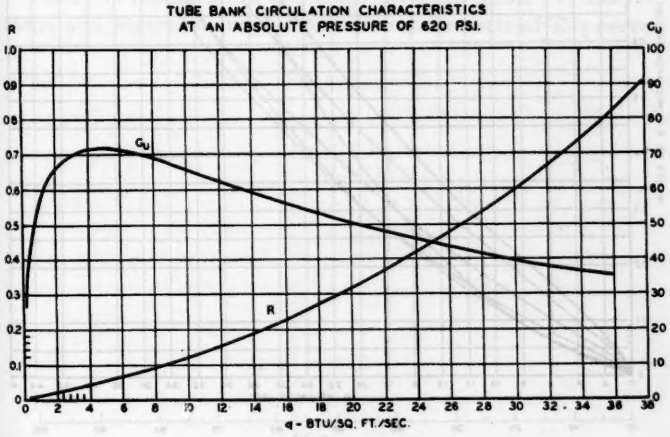


FIGURE 3.

Solution for Five Different Absolute Pressures.

The curves of Figure 3, as was stated in the beginning, can be regarded as applicable to the tube bank of Figure 2 or, merely, as applicable to the two-tube circuit of Figure 1, which has been proportioned to make applicable the constants used. Abandoning consideration of application to tube banks and considering the two-tube solution only, calculations are repeated for additional absolute pressures of 320, 1000, 1500 and 2000 psi without change of any other constants. The resultant set of curves constructed as Figure 4 indicates the R-q relation for five different absolute pressures. For the present the presence of the second abscissae scale "per cent $(q)_m$" in Figures 4 to 8, inclusive, may be disregarded.

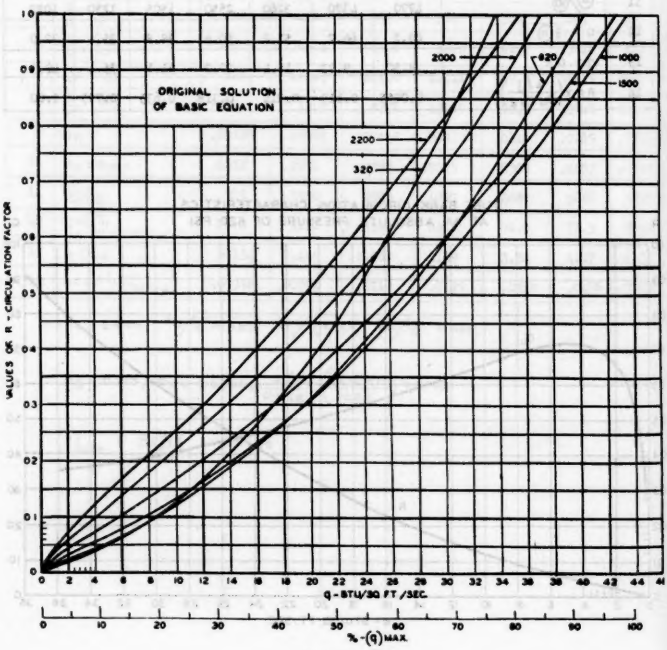


Figure 4.

(First and Second Alternate Solution).

Whatever conclusions may be drawn from an examination of Figure 4 are open to the objection that they fit a special case—a particular set of proportion and flow loss constants. Accordingly, recourse is had to the device of repeating a five-pressure solution for four additional different sets of proportion and flow loss constants in which there is made a systematic controlled variation in values to determine the effect of these changes upon the resultant sets of R-q curves and to determine, if possible, if any type of graphical general solution for the basic equation exists. The first solution, for convenience, is designated as the original solution while each subsequent trial is called a variation or alternate. A description of the four variations is as follows:

First and Second Variations.

As a first trial, the values of the downcomer and baffle entrance and exit loss coefficients are radically reduced (first alternate solution), and then radically increased (second alternate solution), meanwhile attempting to maintain the K_B/K_D ratio reasonably constant (see equations 32 and 42).

Table III indicates the operations involved, Table IV analyzes them, and Figures 5 and 6 depict the resulant sets of R-q curves.

TABLE III—CALCULATIONS FOR FIRST AND SECOND ALTERNATE SOLUTIONS OF BASIC EQUATION.

Variations in K_D and K_B

$$K_B = \left[\left(\frac{\pi D_U^2 N_U}{8 X_Y}\right)^2 \left(1 + {}_rK_1 + \frac{2 f_r l_r (x+y)}{x y}\right) + \left(\frac{N_U}{N_S}\right)^2 \left(\frac{\pi D_U^2}{4 U Z}\right)^2 \left(1 + {}_nK_1 + {}_nK_2\right)\right]$$

Original

$$\left[(9.18)\,(1 + 0.2 + 0.084) + (18.2)\,(1 + 0.2 + 0.2)\right] = 37.3$$

1st Variation

$$\left[(9.18)\,(1 + 0 + 0) + 18.2\,(1 + 0 + 0)\right] = 27.4$$

2nd Variation

$$\left[(9.18)\,(1 + 0.6 + 0.084) + (18.2)\,(1 + 0.6 + 0.6)\right] = 55.4$$

$$K_D = \left[\left(\frac{N_U}{N_d}\right)^2 \left(\frac{D_U}{D_d}\right)^4 \left({}_dK_1 + {}_dK_2 + \frac{4 f_d l_d}{D_d}\right)\right]$$

Original

$$24.8\,(0.2 + 0.2 + 0.907) = 32.4$$

1st Variation

$$24.8\,(0 + 0 + 0.907) = 22.5$$

2nd Variation

$$24.8\,(0.6 + 0.6 + 0.907) = 52.3$$

TABLE III.

$\Psi_A = (v_f/2g)\left[K_D + (K_U)_1 + K_B\right]$

			Ψ_A	P
	Original			
$(2.96/10^4)$	(32.4 + 4.66 + 37.3)		= 0.0220	320
$(3.14/10^4)$			= 0.0234	620
$(3.37/10^4)$		74.4	= 0.0251	1000
$(3.73/10^4)$			= 0.0277	1500
$(4.12/10^4)$			= 0.0306	2000
	1st Variation			
$(2.96/10^4)$	(22.5 + 4.66 + 27.4)		= 0.0162	320
$(3.14/10^4)$			= 0.0172	620
$(3.37/10^4)$		54.6	= 0.0184	1000
$(3.73/10^4)$			= 0.0204	1500
$(4.12/10^4)$			= 0.0225	2000
	2nd Variation			
$(2.96/10^4)$	(55.4 + 4.66 + 52.3)		= 0.0333	320
$(3.14/10^4)$			= 0.0353	620
$(3.37/10^4)$		112.4	= 0.0379	1000
$(3.73/10^4)$			= 0.0419	1500
$(4.12/10^4)$			= 0.0463	2000

TABLE III.

$$\Psi_B = 2\,(\nu_{fg}/g\mu_{fg})(\ell_U/D_U)\left[(\kappa_U)_2 + \kappa_B\right]$$

Original	Ψ_B	P
$2(5.53/10^5)(144)(3.73 + 37.3)$	= 0.654	320
$(3.09/10^5)$ " "	= 0.365	620
$(2.03/10^5)$ " "	= 0.240	1000
$(1.42/10^5)$ " "	= 0.168	1500
$(1.09/10^5)$ " "	= 0.129	2000
1st Variation		
$2(5.53/10^5)(144)(3.73 + 27.4)$	= 0.495	320
$(3.09/10^5)$ " "	= 0.276	620
$(2.03/10^5)$ " "	= 0.182	1000
$(1.42/10^5)$ " "	= 0.127	1500
$(1.09/10^5)$ " "	= 0.0975	2000
2nd Variation		
$2(5.53/10^5)(144)(3.73 + 55.4)$	= 0.940	320
$(3.09/10^5)$ " "	= 0.525	620
$(2.03/10^5)$ " "	= 0.345	1000
$(1.42/10^5)$ " "	= 0.241	1500
$(1.09/10^5)$ " "	= 0.185	2000

TABLE IV.—SUMMARY OF CHANGES IN CONSTANTS INVOLVED IN THE FIRST AND SECOND ALTERNATE SOLUTIONS OF BASIC EQUATION.

	1st variation	original	2nd variation
$_{A}K_{1}$	0	0.2	0.6
f_A	0	0.006	0.006
$_{h}K_{1}$	0	0.2	0.6
$_{h}K_{2}$	0	0.2	0.6
K_B	27.4	37.3	55.4
$_{d}K_{1}$	0	0.2	0.6
$_{d}K_{2}$	0	0.2	0.6
f_d	0.006	0.006	0.006
K_D	22.5	32.4	52.3
K_B/K_D	1.22	1.15	1.06
(a) $\Psi_A/(V_f/2g)$	54.6	74.4	112.4
(b) $\Psi_B/2(V_{fg}/_{g}H_{fg})(\ell_u/D_u)$	31.1	41.0	59.1
(a)/(b)	1.76	1.81	1.90
ΔK_B, %	-26.5		+48.6
ΔK_D, %	-30.6		+30.6
Δ (a), %	-26.6		+51.2
Δ (b), %	-24.1		+44.2

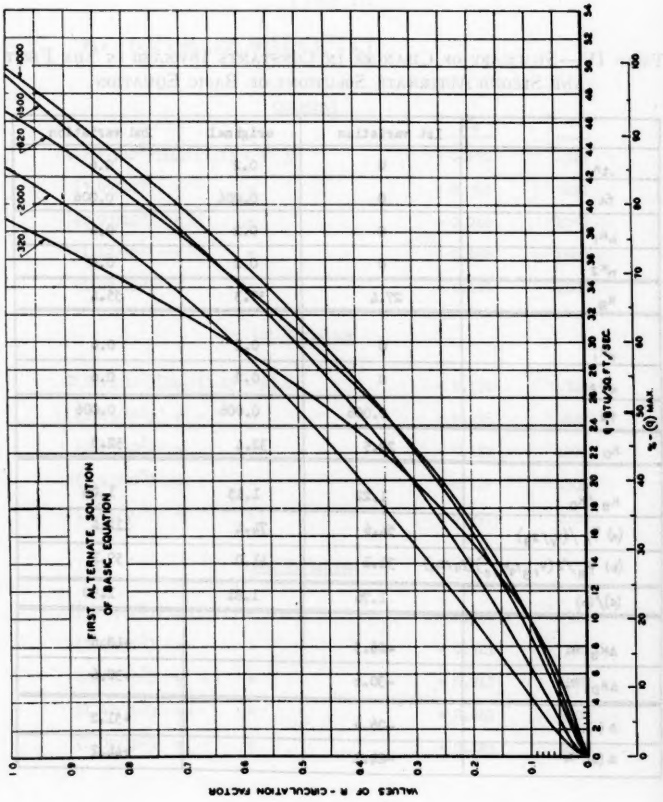


FIGURE 5.

VALUES OF R - CIRCULATION FACTOR

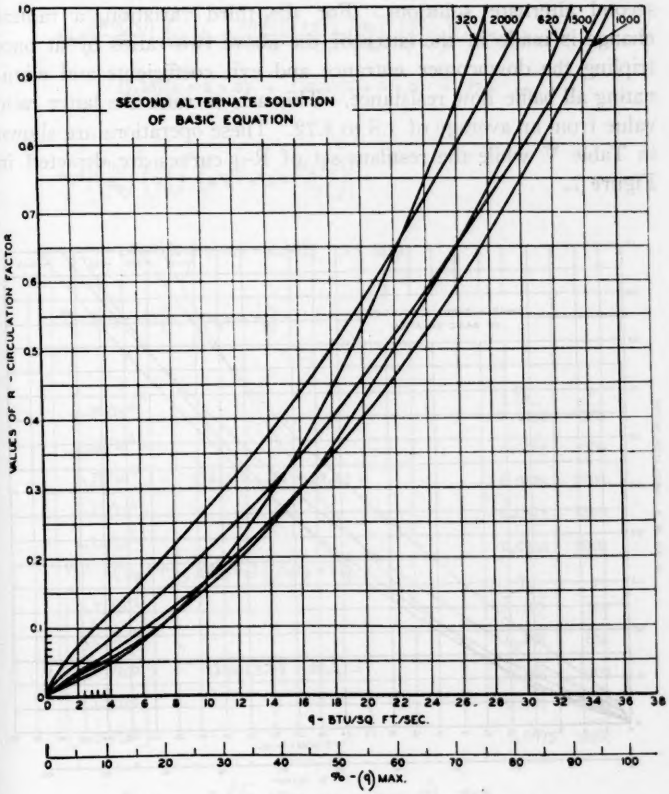

SECOND ALTERNATE SOLUTION
OF BASIC EQUATION
320
2000
820
1500
1000
VALUES OF R - CIRCULATION FACTOR
q - BTU/SQ. FT./SEC.
% - (q) MAX.

FIGURE 6.

Third Alternate Solution.

It will be noted in Table IV that both the (K_B/K_D) and the $[K_D + (K_u)_1 + K_B]/[(K_u)_2 + K_B]$ ratios (see equation (47)) were maintained approximately constant for the original, first, and second alternate solutions. For the third variation, a radical change is made in the latter of the above two ratios by at once tripling the downcomer entrance and exit coefficients and eliminating all baffle flow resistance. This acts to raise the latter ratio value from an average of 1.8 to 2.72. These operations are shown in Table V while the resulant set of R-q curves are depicted in Figure 7.

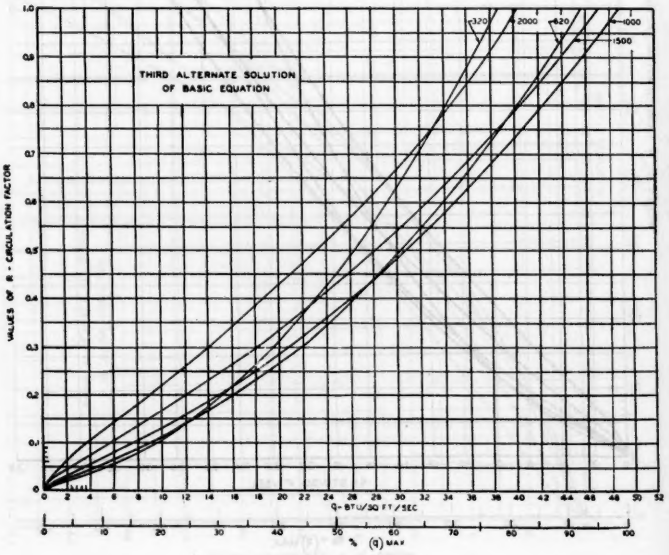


Figure 7.

TABLE V—CALCULATIONS FOR THIRD ALTERNATE SOLUTION.

$$K_B = \left[\left(\frac{\pi D_U^2 N_U}{8xy}\right)^2 \left(1 + pK_1 + \frac{2f_r \ell_r (x+y)}{xy}\right) + \left(\frac{N_U}{N_S}\right)^2 \left(\frac{\pi D_U^2}{4UZ}\right)^2 (1 + nK_1 + nK_2)\right] =$$

$$(9.18)(1 + 0 + 0) + (18.2)(1 + 0 + 0) = 27.4$$

$$K_D = \left[\left(\frac{N_U}{N_D}\right)^2 \left(\frac{D_U}{D_d}\right)^4 \left(dK_1 + dK_2 + \frac{4f_d \ell_d}{D_d}\right)\right] =$$

$$(24.8)(0.6 + 0.6 + 0.907) = 52.3$$

$$\nabla_A = (v_{fg}/2g)\left[K_D + (K_U)_1 + K_B\right]$$

		∇_A	P
$2.96/10^4$		0.0250	320
$3.14/10^4$		0.0265	620
$3.37/10^4$	$(52.3 + 4.66 + 27.4) =$	0.0284	1000
$3.73/10^4$		0.0315	1500
$4.12/10^4$		0.0348	2000

$$\nabla_B = 2(v_{fg}/gH_{fg})(\ell_U/D_U)\left[(K_U)_2 + K_B\right]$$

		∇_B	P
$5.53/10^5$		0.495	320
$3.09/10^5$		0.276	620
$2.03/10^5$	$(144)(3.73 + 27.4) =$	0.182	1000
$1.42/10^5$		0.127	1500
$1.09/10^5$		0.0975	2000

Here, $\left[\nabla_A/(v_4/2g)\right] \Big/ \left(\nabla_B/\left[2(v_{fg}/gH_{fg})(\ell/D)\right]\right) = \frac{84.4}{31.1} = 2.72$ as compared with the ratio 1.8 (approximate) previously used.

Fourth Alternate Solution.

As a fourth variation, flow loss coefficient values are restored to their original magnitudes while the upcomer and downcomer lengths (only) are halved. The new elevations and lengths are:

$$(Z_h)_2 = (Z_h - Z_u)_1 + \tfrac{1}{2}(Z_u)_1$$
$$(Z_u)_2 = \tfrac{1}{2}(Z_u)_1$$
$$(l_u)_2 = \tfrac{1}{2}(l_u)_1$$
$$(l_d)_2 = \tfrac{1}{2}(l_d)_1$$

Table VI summarizes the constant values which result while Figure 8 portrays the resultant set of R-q curves.

Table VI—Summary of Constants.

z_h	= 6.2 ft.	U	= 2.5/12 ft.
z_U	= 4.9 ft.	z	= 10.5/12 ft.
z_r	= 1.8 ft.		
l_U	= 6 ft.		
D_U	= 1/12 ft.	$_dK_1$	= 0.2
D_L	= 2/12 ft.	$_dK_2$	= 0.2
N_d	= 1805	$_UK_1$	= 0.2
N_L	= 56	$_rK_1$	= 0.2
N_U	= 2029	ηK_1	= 0.2
l_d	= 6 ft.	ηK_2	= 0.2
D_d	= 3.814/12 ft.	f_d	= 0.006
N_d	= 28	f_U	= 0.006
N_{ST}	= 22	f_r	= 0.006
N_S	= 14.3	θ_5	= % steam generated in large tube bank
y_T	= 126.5/12		
x	= 3.2/12		
y	= 82.2/12 ft.		

TABLE VI.

$$\Psi_A$$

$$K_D = \left[\left(\frac{N_U}{N_d}\right)^2 \left(\frac{D_U}{D_d}\right)^4 \left(dK_1 + dK_2 + \frac{4 f_d l_d}{D_d}\right)\right]$$

$$= 24.8 \left(0.2 + 0.2 + 0.454\right) = \quad 20.95$$

$$(K_U)_1 = \left(1 + UK_1 + 4\frac{f_U l_U}{D_U}\right) = \left(1 + 0.2 + 1.73\right) = \quad 2.93$$

$$K_B = \left[\left(\frac{\pi D_0^2 N_U}{8 x y}\right)^2 \left(1 + {}_pK_1 + \frac{2 f_r l_r (x+y)}{x y}\right) + \left(\frac{N_U}{N_B}\right)^2 \left(\frac{\pi D_0^2}{4 U Z}\right)^2 \left(1 + {}_pK_1 + {}_pK_2\right)\right] =$$

$$\left(9.18\right)\left(1 + 0.2 + 0.084\right) + \left(18.2\right)\left(1 + 0.2 + 0.2\right) = \quad 37.3$$

$$\Psi_A = \frac{V_f}{2g}\left[K_D + (K_U)_1 + K_B\right]$$

		Ψ_A	P
$2.98/10^4$	$(21.0 + 2.9 + 37.3) =$	0.0181	320
$3.14/10^4$		0.0192	620
$3.37/10^4$	61.2	0.0206	1000
$3.73/10^4$		0.0228	1500
$4.12/10^4$		0.0252	2000

$$\Psi_B$$

$$(K_U)_2 = \tfrac{1}{2}\left(4 f_U l_U / D_U + 4\right) = \tfrac{1}{2}\left(1.73 + 4\right) = \quad 2.83$$

$$\Psi_B = 2\left(V_f g / g H_f g\right)\left(f_U / D_U\right)\left[(K_U)_2 + K_B\right]$$

		Ψ_B	P
$5.53/10^5$	$72\ (2.83 + 37.3) =$	0.319	320
$3.09/10^5$		0.178	620
$2.03/10^5$	401	0.117	1000
$1.42/10^5$		0.0818	1500
$1.09/10^5$		0.0628	2000

TABLE VI.

Absolute Pressure	320	620	1000	1500	2000
z_h/v_f	326	307	286	259	234
z_U/v_f	258	243	226	205	185
z_r/v_f	94.6	89.1	82.9	75.3	67.8
ϕ_I	26.9	14.23	8.70	5.50	3.80
$v_f/2g$	$\frac{2.96}{10^4}$	$\frac{3.14}{10^4}$	$\frac{3.37}{10^4}$	$\frac{3.71}{10^4}$	$\frac{4.12}{10^4}$
$2\left(\frac{v_{fg}}{gH_{fg}}\right)\left(\frac{l_U}{D_U}\right)$	0.0796	0.00446	0.0292	0.0204	0.0157
N_U	2029	2029	2029	2029	2029
$\left(\frac{N_U}{N_d}\right)^2\left(\frac{D_U}{D_d}\right)^4$	24.8	24.8	24.8	24.8	24.8
$\left(\frac{4 f_d l_d}{D_d}\right)$	0.454	0.454	0.454	0.454	0.454
y	82.2/12	82.2/12	82.2/12	82.2/12	82.2/12
$\left(\frac{\pi D_U^2 N_U}{8 x_y}\right)^2$	9.18	9.18	9.18	9.18	9.18
$\frac{2 f_r l_r}{x_y}(x+y)$	0.084	0.084	0.084	0.084	0.084
N_S	14.3	14.3	14.3	14.3	14.3
$\left(\frac{N_U}{N_S}\right)^2\left(\frac{\pi D_U^2}{4UZ}\right)^2$	18.2	18.2	18.2	18.2	18.2
$4\frac{f_U l_U}{D_U}$	1.73	1.73	1.73	1.73	1.73
$4 l_U/D_U H_{fg}$	0.359	0.397	0.448	0.523	0.626
$(x)_{MAX.} = \left(\frac{D_U H_{fg}}{4 l_U}\right)$	2.79	2.52	2.23	1.91	1.60

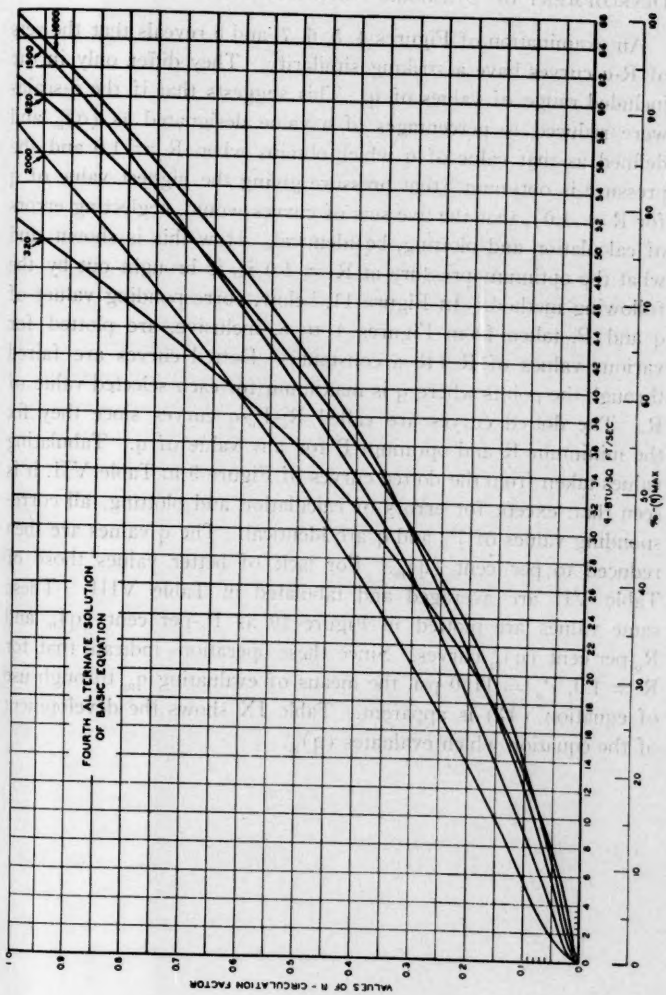

FOURTH ALTERNATE SOLUTION
OF BASIC EQUATION
VALUES OF R - CIRCULATION FACTOR
Q - BTU/SQ FT/SEC
θ_{max}

FIGURE 8.

Development of Standard Solution for Two-Tube Circuit.

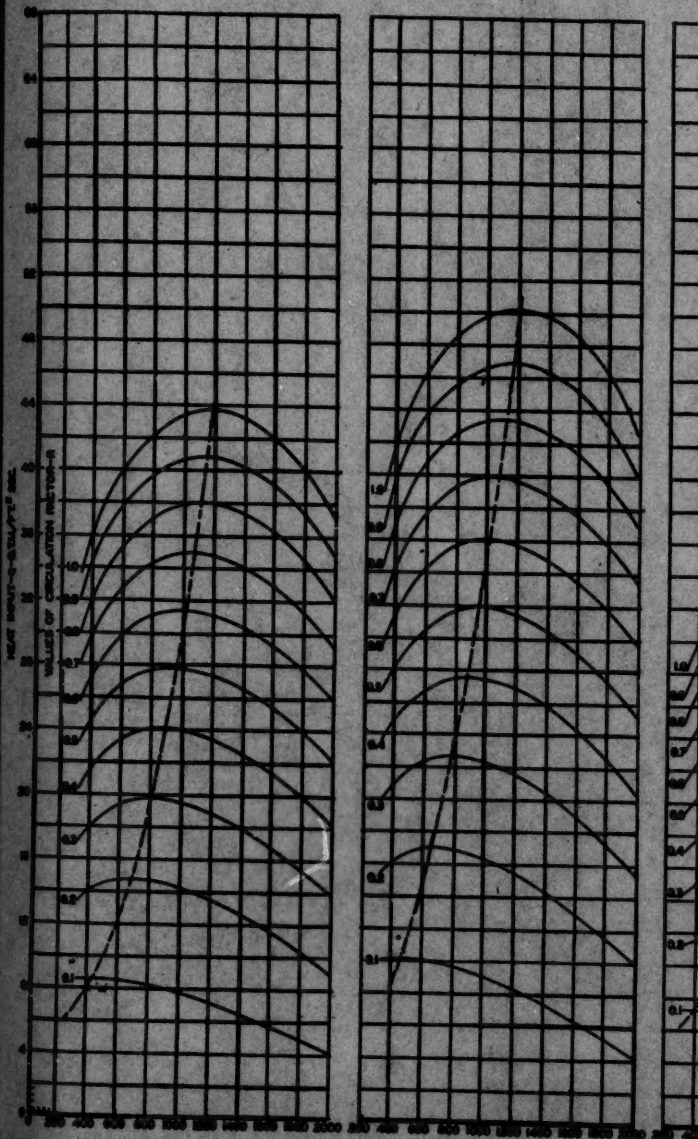
An examination of Figures 4, 5, 6, 7, and 8 reveals that the sets of R-q curves have a striking similarity. They differ only in the included range of values of q. This suggests that if the q-scales were reduced to percentages of a value designated as $(q)_m$ and defined as that value of q which obtains when $R = 1.0$ and the pressure is optimum (that pressure giving the highest value of q for $R = 1.0$), that the five sets of curves would, neglecting errors of calculation and plotting, be identical. How this is shown, and what the optimum pressure at $R = 1.0$ is, is brought out by the following method. In Figure 11, below, corresponding values of q and P, taken from Figures 4 to 8, inclusive, are plotted for various values of R (R a constant). Dotted curves are faired through the points where q is maximum for each selected value of R. The dotted curves are called R_o-P_o-q curves since they fix the minimum R and optimum P for any value of q. Tabulating values taken from the dotted curves of Figure 9 in Table VII, it is seen that, except for errors of calculation and plotting, all corresponding values of P_o and q are identical. The q values are then reduced to per cent $(q)_m$. For lack of better values those of Table VII are averaged and tabulated in Table VIII. These same values are plotted in Figure 10 as P_o-per cent $(q)_m$ and R_o-per cent $(q)_m$ curves. Since these operations indicate that for $R = 1.0$, $P_o = 1150$ psi, the means of evaluating q_m through use of equation (47) is apparent. Table IX shows the development of the equation which evaluates $(q)_m$.

(TAKEN FROM FIG. 4)

ORIGINAL SOLUTION
WITH
NORMAL FLOW RESISTANCES

(TAKEN FROM FIG. 5)

FIRST ALTERNATE SOLUTION
WITH
REDUCED FLOW RESISTANCES



$R-P^2_q$ RELATION

(TAKEN FROM FIG. 6)

SECOND ALTERNATE SOLUTION WITH INCREASED FLOW RESISTANCES

(TAKEN FROM FIG. 7)

THIRD ALTERNATE SOLUTION WITH REDUCED BAFFLE RESISTANCE AND INCREASED DOWNCOMER RESISTANCE

(TAKEN FROM FIG. 8)

FOURTH ALTERNATE SOLUTION WITH TUBE BANK HEIGHT REDUCED BY ONE HALF FLOW RESISTANCE AT NORMAL VALUES

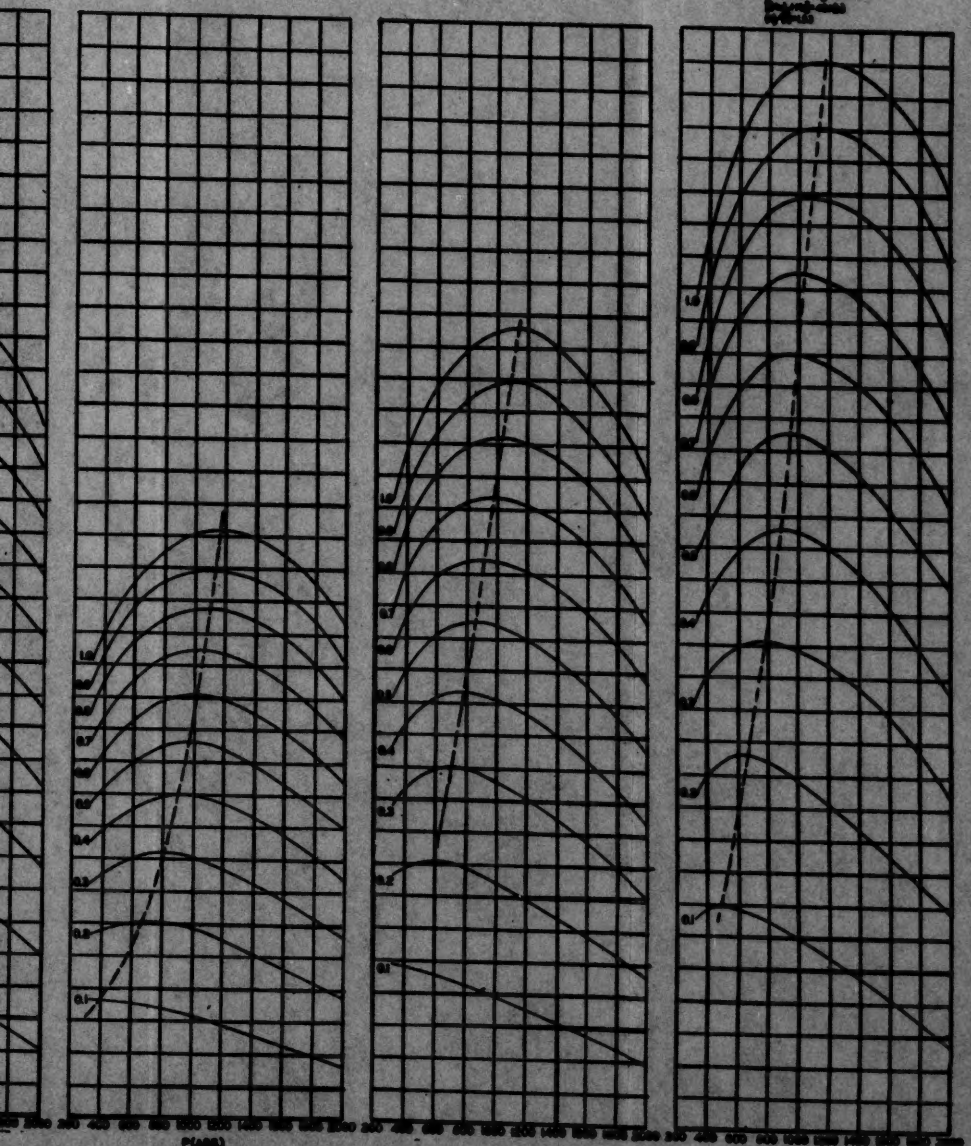


FIGURE 9.

TABLE VII—TABULATED VALUES OF P_o VS Q AND PER CENT (QM) FOR SELECTED VALUES OF R TAKEN FROM FIGURE 9.

CASE	R	P_o	q	% (q) MAX.	R	P_o	q	% (q) MAX.
1	1.0	1175	43.7	100	0.5	930	27.8	63.7
2	1.0	1210	50.1	100	0.5	950	31.9	63.5
3	1.0	1175	36.3	100	0.5	970	23.2	63.9
4	1.0	1150	49.2	100	0.5	890	31.0	63.0
5	1.0	1160	66.0	100	0.5	950	43.0	65.2
1	0.9	1150	41.8	96.0	0.4	870	24.1	55.2
2	0.9	1170	46.9	93.5	0.4	860	27.5	54.9
3	0.9	1140	33.8	93.0	0.4	930	20.0	55.2
4	0.9	1120	45.8	93.2	0.4	780	26.6	54.1
5	0.9	1130	62.0	94.0	0.4	880	37.2	56.3
1	0.8	1110	38.0	87.0	0.3	800	19.7	45.2
2	0.8	1080	43.4	86.5	0.3	770	22.6	45.1
3	0.8	1100	31.5	87.0	0.3	800	16.3	45.1
4	0.8	1030	42.3	86.2	0.3	700	22.0	44.7
5	0.8	1060	57.7	87.2	0.3	780	30.2	45.7
1	0.7	1050	35.0	80.0	0.2	680	14.8	33.9
2	0.7	1020	39.9	79.5	0.2	670	17.0	33.9
3	0.7	1030	28.9	79.6	0.2	670	12.0	33.1
4	0.7	970	38.6	78.5	0.2	580	16.2	33.0
5	0.7	970	53.0	80.2	0.2	646	23.2	35.1
1	0.6	990	31.3	72.0	0.1	440	8.5	19.5
2	0.6	980	36.1	72.0	0.1	460	10.1	20.1
3	0.6	1000	26.1	72.0	0.1	400	7.3	20.1
4	0.6	900	34.7	70.5	0.1	500	9.4	19.1
5	0.6	960	48.0	72.8	0.1	500	14.0	21.2

TABLE VIII—TABULATION OF AVERAGED VALUES OF TABLE VII.

R_o	P_o	% (q) MAX.
1.0	1150	100
0.9	1142	93.9
0.8	1074	86.8
0.7	1008	79.6
0.6	966	71.9
0.5	938	63.9
0.4	864	55.1
0.3	770	45.2
0.2	648	33.8
0.1	460	20.0

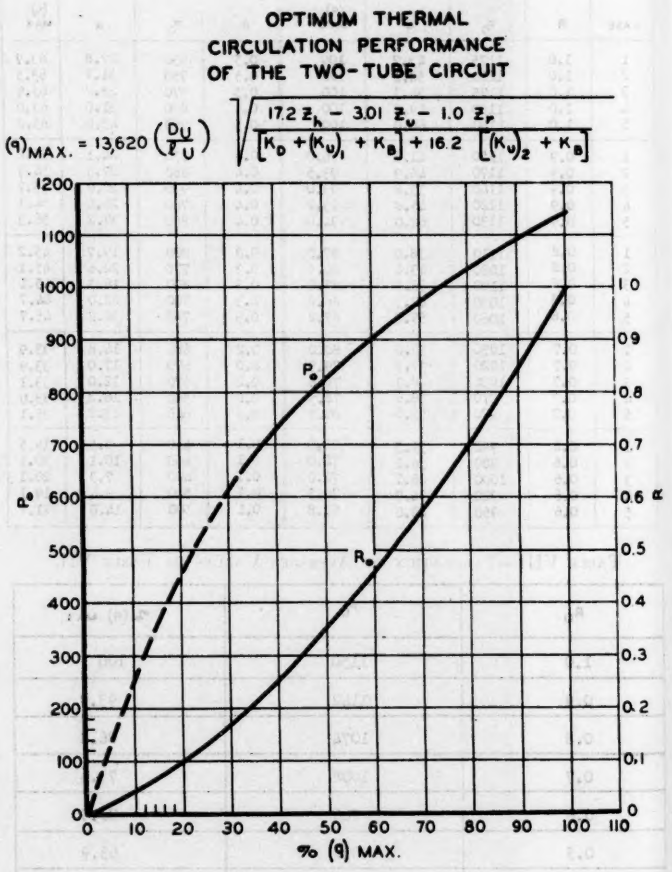

OPTIMUM THERMAL
CIRCULATION PERFORMANCE
OF THE TWO-TUBE CIRCUIT
$(q)_{MAX.} = 13{,}620 \left(\frac{D_U}{\ell_U}\right) \sqrt{\frac{17.2\,\bar{z}_h - 3.01\,\bar{z}_v - 1.0\,\bar{z}_r}{[K_D + (K_v)_1 + K_B] + 16.2\,[(K_v)_2 + K_B]}}$
P_o
R_o
α°
α
% (q) MAX.

FIGURE 10.

TABLE IX.

$$G_U = \sqrt{\frac{z_h/v_f - \left(z_U/v_f\right)\dfrac{\ell_n(1+v_{fg}/v_f\,R)}{\phi_I\,R} - \dfrac{z_r}{v_f\,(1+\phi_I R)}}{\dfrac{v_f}{2g}\left[K_D + (K_U)_1 + K_B\right] + \dfrac{v_{fg}}{2g}\left[(K_U)_2 + K_B\right]R}}, \quad \text{WHERE } \phi_I = \left(\frac{4\,\ell_U\,v_{fg}}{D_U H_{fg}\,v_f}\right) \tag{47}$$

$$R = \left(\frac{4\,\ell_U}{D_U H_{fg}}\right) \quad X,\ X = R\left(\frac{D_U H_{fg}}{4\,\ell_U}\right) = \left(\frac{q}{G_U}\right),\ G_U = \left(\frac{q}{R}\right)\left(\frac{4\,\ell_U}{D_U H_{fg}}\right)$$

$$\phi_I\,x = 4\left(\frac{\ell_U}{D_U}\right)\left(\frac{v_{fg}}{v_f H_{fg}}\right)R\left(\frac{D_U H_{fg}}{4\,\ell_U}\right) = R\left(\frac{v_{fg}}{v_f}\right)$$

$$2\left(\frac{v_{fg}}{g H_{fg}}\right)\left(\frac{\ell_U}{D_U}\right)x = 2\left(\frac{v_{fg}}{g H_{fg}}\right)\left(\frac{\ell_U}{D_U}\right)R\left(\frac{D_U H_{fg}}{4\,\ell_U}\right) = \left(\frac{v_{fg}}{2g}\right)R$$

THEN

$$q = R\left(\frac{D_U H_{fg}}{4\,\ell_U}\right)\sqrt{\frac{\dfrac{z_h}{v_f} - \dfrac{z_U}{v_{fg}R}\,\ell_n\left(1 + v_{fg}/v_f\,R\right) - \dfrac{z_r}{(v_f + v_{fg}R)}}{\dfrac{v_f}{2g}\left[K_D + (K_U)_1 + K_B\right] + \dfrac{v_{fg}}{2g}\left[(K_U)_2 + K_B\right]R}} \tag{48}$$

IF P = 1150 AND R = 1.0

$$\left(\frac{R\,H_{fg}}{4}\right) = \frac{1.0 \times 621}{4} = 155.5,\ \frac{1}{v_f} = 45.2,\ \frac{1}{v_{fg}} = 2.79$$

$$\frac{v_{fg}}{v_f} = 16.2,\ \ell_n\left(1+16.2\right) = 2.84,\ 2.79 \times 2.84 = 7.93$$

$$\frac{1}{v_f + v_{fg}} = \frac{1}{0.3802} = 2.63,\ \frac{v_f}{2g} = \frac{0.0221}{64.4} = \frac{3.43}{10^4}$$

$$\frac{v_{fg}}{2g} = \frac{0.3581}{64.4} = \frac{55.7}{10^4}$$

See footnote on next page.

TABLE IX.

$$(q)_{MAX} = 155.5 \left(\frac{D_U}{\ell_U}\right) \sqrt{\frac{45.2\ z_h - 7.93\ z_U - 2.63\ z_r}{\frac{3.43}{10^4}\left[\kappa_D + (\kappa_U)_1 + \kappa_B\right] + \frac{55.7}{10^4}\left[(\kappa_U)_2 + \kappa_B\right]}} \qquad (48)$$

or

$$\text{MAX.} = 13{,}620 \left(\frac{D_U}{\ell_U}\right) \sqrt{\frac{17.2\ z_h - 3.01\ z_U - 1.0\ z_r}{\left[\kappa_D + (\kappa_U)_1 + \kappa_B\right] + 16.2\left[(\kappa_U)_2 + \kappa_B\right]}} \qquad (49)$$

Equation (49) evaluates q_{MAX}, the value of q corresponding to R=1.0 and P = 1150 psi.

It now remains to be shown that, not only may the R_o-P_o-q curves (dotted curves of Figure 9) for each solution be reduced to a common R_o-P_o-per cent (q) curve (see Figure 10), but also, that all R-q curves (for constant values of P) will reduce to common R-per cent $(q)_m$ (P a constant) curves. Per cent $(q)_m$ scales are added to Figures 4 to 8, inclusive. Corresponding

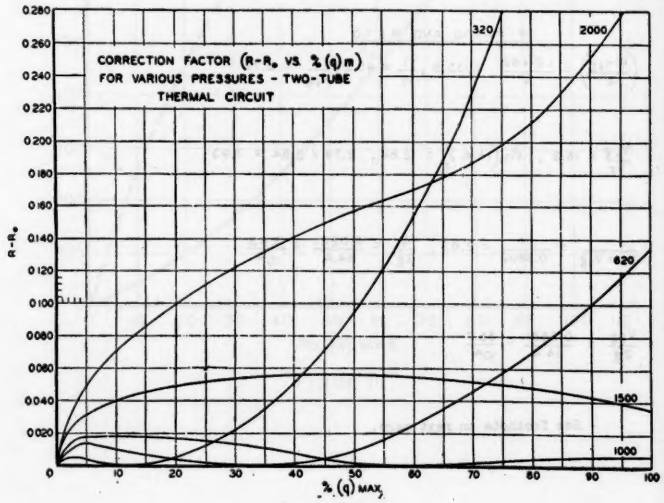


FIGURE 11.

values of R, Ro, and (R-Ro), for selected values of per cent $(q)_m$, are taken from Figures 4 to 8, inclusive and Figure 10, and tabulated for comparison. It is seen that except for errors of calculation and plotting, (R-Ro) vs per cent $(q)_m$ is also a common relation at each selected pressure. These are shown in Figure 11 in the form of correction factors to the R-per cent $(q)_m$ curve of Figure 10, although they have no particular utility.

Conclusions.

If, now, pressures were selected say, in 100 psi increments from 0 to the critical pressure, and R-per cent $(q)_m$ curves plotted, there would result a diagram which would reveal full thermal circulation information regarding any two-tube (or U-tube) circuit provided only, that the value of $(q)_m$ were known. $(q)_m$ in turn may be readily evaluated by use of equation (49), an operation involving only evaluation of circuit flow resistance coefficients and dimensions. The form of equations (32), (42), (45), and (46) suggests that this would be a simple operation. The necessity for this single operation in turn suggests the idea of utilizing $(q)_m$ as an index or rating for a circuit since it is not only a key to the complete solution, but also by its relative magnitude a direct index of the circuit capacity to sustain high heat input rates. The values of $(q)_m$ for the five cases or solutions are, in their numerical order, 43.8, 50.2, 36.2, 48.6, and 65.7. If by experience standard values of this index are established, then boiler designers might even dispense with the R-q diagram and judge the predicted thermal circulation performance of a circuit on the basis of the calculated value of $(q)_m$ alone.

Application.

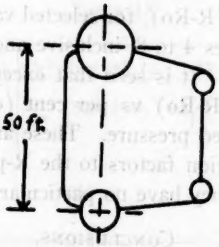
Taking an example from Markson, Ravese, and Humphreys' work, a trial application is made as follows:

DATA FOR STEAM BOILER FURNACE CIRCUIT.

Risers—161 in number, 3-inch O.D., 0.198 ft. I.D., developed length 90 ft., projected heating surface 2700, vertical height 50 ft.

Downcomers—Two 12″ I.D., developed length 60 ft.

Pressure—2200 psi absolute.



The assumed nominal rate of heat absorption is 49,000 Btu. per foot of projected area per hour. Each tube absorbs the same quantity of heat.

Repeating Equation (49):

$$(q)_m = 13{,}620\left(\frac{D_u}{l_u}\right)\sqrt{\frac{17.2Z_h - 3.01\,Z_u - 1.0\,Z_r}{[K_D+(K_u)_1+K_B]+16.2[(K_u)_2+(K_B]}}$$

$D_u/l_u = 0.198/90 = 1/454$, $17.2\,Z_h = 17.2 \times 50 = 860$, $3.01\,Z_u = 3.01 \times 45 = 135.5$, $Z_r = 2.5$

$$K_D = \left[\frac{(161/2)^2(0.198/1.0)^4(0.2+0.2+4\times6\times60/1\times1000)}{(N_u/N_d)^2\,(D_u/D_d)^4\,({}_dK_1 + {}_dK_2 + 4f_d l_d/D_d)}\right] = 18.3$$

$$K_B = \left[\frac{(1)\qquad (1)\quad (1+0.5+2\times0.2)}{(N_u/N_s)(\pi D_u^2/4_{us})^2(1+{}_nK_1+{}_nK_2)}\right] = 1.9$$

$$(K_u)_1 = \left[\frac{1+0.2+4\times6\times90/1000\times0.198}{1+{}_uK_1+4f_u l_u/D_u}\right] = 12.1$$

$$(K_u)_2 = \left[\frac{\frac{1}{2}\,(4\times6\times90/1000\times0.198+4}{\frac{1}{2}\,(4f_u l_u/D_u+4}\right] = 7.5$$

$$(q)_m = \left(\frac{13.620}{454}\right)\sqrt{\frac{722}{184.6}} = 59.4\ (\text{Btu}/\text{ft.}^2\text{sec})$$

$$\%(q)_m = \frac{\left.\begin{matrix}133\\524\\769\\1160\end{matrix}\right\} H(2)\dagger \times 10^6 \times 100}{60 \times 60 \times 161 \times 90 \times \pi \times 0.198 \times 59.4} =$$

$\%(q)_m$	R(1)*	R(2)
6.9	0.10	0.11
27.1	0.29	0.35
40.0	0.44	0.50
60.2	0.71	0.75

* See Figure 4. † See following explanation.

(1) By methods outlined in this work.

(2) From example by Markson, Ravese, and Humphreys.

For the above example, an R-q curve for 2200 psi was calculated, using constants given in the previous sample solution, and added to Figure 4. Using the per cent $(q)_m$ scale, previously added to this Figure, values of R (marked *) were taken for values of per cent $(q)_m$ which corresponded to values of H (total heat to circuit) which were calculated by Markson, Ravese, and Humphreys for assumed values for R of 0.11, 0.35, 0.50, and 0.75. Equation (49) and preceding ones do not make provision for intermediate entrance and exit losses (headers) in the upcomer leg. A decision was made that locating one header near the upcomer leg exit would suitably represent the presence of two headers located approximately 1/3 and 2/3 the total developed upcomer length from the upcomer entrance. This, in effect, was accomplished by including in K_B the term " $+ 2 \times 0.2$ ". Since friction losses are predominant in the above example, no serious error results from this approximation. It will be noted that, by methods given in this work, the evaluation of $(q)_m$ for a circuit, makes immediately available without further effort, any given number of corresponding values of R and per cent $(q)_m$, and for any selected pressure, provided only, that a set of R-per cent $(q)_m$ curves is available.

ENTHALPY OF SUPERHEATED STEAM.

By James W. Murdock.*

The ordinary steam tables necessitate laborious interpolations to determine the enthalpy or total heat of superheated steam with any degree of accuracy. The accompanying table has been compiled to reduce interpolation to two simple calculations and to present a wide range of values on a single sheet. The values of enthalpy are given in Btu. per pound of steam in increments of one hundred pounds per square inch pressure, and ten degrees temperature (F). It is believed that the wide range of pressures and temperatures selected will make the table useful to Naval personnel concerned with boilers.

The table is constructed in the following manner: The pressures are extended horizontally across the top of the page and between each value of pressure is a column marked " d ", in which the value shown represents an increment of enthalpy for one pound increment of pressure between these pressures. In a similar manner the temperatures are extended vertically and between each line of temperature is a line marked " d ", which represents an increment of enthalpy for one degree increment of temperature between these temperatures. These increments of enthalpy represent the average for a ten degree range of temperature and a hundred pound range of pressure.

Values of enthalpy are given to the nearest hundredth of a Btu./lb. instead of the usual tenth. This is done to avoid a possible cumulative error when using some of the values which had to be interpolated from the original steam tables. It is to be noted that the use of this table may yield answers slightly different from those obtained when interpolating from the steam tables in the ordinary manner. This difference, however, in no instance is

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PRESSURE - P.S.I. - ABS.

TOTAL HEAT OF SUPERHEATED STEAM

	100	d	200	d	300	d	400	d	500	d	600	d	700	d	800	d	900	d	1000
400	1227.60	.1730	1220.30																
d	.525		.630																
410	1232.85	.1625	1216.60																
d	.525		.630																
420	1238.10	.1530	1222.90	.1810	1204.80														
d	.525		.600		.725														
430	1243.35	.1445	1228.90	.1695	1211.75														
d	.525		.600		.715														
440	1248.60	.1370	1234.90	.1580	1219.10														
d	.510		.580		.670														
450	1253.70	.1300	1240.70	.1490	1225.80	.1700	1208.80												
d	.510		.580		.670		.770												
460	1258.80	.1230	1246.30	.1400	1232.50	.1600	1216.50												
d	.510		.565		.640		.750												
470	1263.90	.1175	1252.15	.1325	1238.90	.1490	1224.00	.1700	1207.00										
d	.510		.565		.640		.730		.840										
480	1269.00	.1120	1257.80	.1250	1245.30	.1400	1231.30	.1990	1215.40										
d	.505		.555		.615		.700		.810										
490	1274.05	.1170	1263.35	.1190	1251.45	.1315	1238.30	.1680	1223.50	.1680	1206.70								
d	.505		.555		.615		.680		.780		.900								
500	1279.10	.1080	1268.90	.1130	1257.60	.1250	1245.10	.1380	1231.30	.1540	1215.70								
d	.505		.545		.595		.660		.735		.860								
510	1284.15	.0980	1274.35	.1080	1263.55	.1185	1251.70	.1305	1238.65	.1435	1224.30	.1630	1208.00						
d	.505		.545		.595		.660		.735		.820		.990						
520	1289.20	.0940	1279.80	.1030	1269.50	.1120	1258.30	.1230	1216.00	.1350	1232.50	.1930	1217.50	.1700	1200.50				
d	.500		.535		.580		.635		.700		.790		.890		1.030				
530	1294.20	.0905	1285.15	.0985	1275.30	.1065	1264.65	.1165	1253.00	.1260	1240.40	.1400	1226.40	.1560	1210.60				
d	.500		.535		.580		.635		.700		.770		.860		.970				
540	1299.20	.0870	1290.30	.0940	1281.10	.1010	1271.00	.1100	1260.00	.1190	1248.10	.1310	1235.00	.1450	1220.50	.1610	1204.40		
d	.500		.530		.570		.615		.670		.760		.820		.930		1.060		
550	1304.20	.0840	1295.80	.0900	1286.80	.0965	1277.15	.1045	1266.70	.1120	1255.50	.1230	1243.20	.1360	1229.80	.1480	1215.00	.1670	1198.30
d	.500		.530		.570		.615		.670		.720		.790		.880		.990		1.140
560	1309.20	.0810	1301.10	.0860	1292.50	.0920	1283.30	.0990	1273.40	.1070	1262.70	.1160	1251.10	.1250	1238.60	.1370	1224.90	.1520	1209.70
d	.500		.530		.560		.595		.640		.700		.770		.840		.940		1.060
570	1314.20	.0780	1306.40	.0830	1298.10	.0885	1289.25	.0945	1279.80	.1010	1269.70	.1090	1258.80	.1180	1247.00	.1270	1234.30	.1400	1220.30
d	.500		.530		.560		.595		.640		.670		.730		.820		.890		1.000
580	1319.20	.0750	1311.70	.0800	1303.70	.0850	1295.20	.0900	1286.20	.0960	1276.60	.1030	1266.30	.1110	1255.20	.1200	1243.20	.1290	1230.30
d	.495		.520		.550		.585		.620		.670		.720		.770		.860		.930
590	1324.15	.0725	1316.90	.0770	1309.20	.0815	1301.05	.0865	1292.40	.0910	1283.30	.0980	1273.50	.1040	1263.10	.1130	1251.80	.1200	1239.80
d	.495		.520		.550		.585		.620		.660		.710		.760		.830		.900
600	1329.10	.0700	1322.10	.0740	1314.70	.0780	1306.90	.0830	1298.60	.0870	1289.90	.0930	1280.60	.0990	1270.70	.1060	1260.10	.1130	1248.80
d	.495		.520		.550		.585		.620		.660		.710		.760		.830		.900

TEMPERATURE - °F

570																					
d		.500																			
580	1319.20		.0790	1311.70	.530	.0800	1303.70	.530	.0850	1295.20	.595	.0900	1286.20	.640	.0960	1276.60	.670	.1030	1266.30	.720	.1110	1255.20	.790	.1200	1243.20	.860	.1290	1230.30	.950
d		.495			.530			.530			.585			.630			.660			.710			.830			.900		
590	1324.15		.0725	1316.90		.0770	1309.20		.0815	1301.05		.0865	1292.40		.0910	1283.30		.0980	1273.50		.1040	1263.10		.1130	1251.80		.1200	1239.80	
d		.495			.530			.530			.585			.630			.660			.710			.830			.900		
600	1329.10		.0700	1322.10		.0740	1314.70		.0780	1306.90		.0830	1298.60		.0870	1289.90		.0930	1280.60		.0990	1270.70		.1060	1260.10		.1130	1248.80	
d		.498			.530			.545			.570			.605			.640			.685			.739			.799			.893
610	1334.08		.0678	1327.30		.0715	1320.13		.0755	1312.60		.0795	1304.65		.0835	1296.30		.0885	1287.45		.0940	1278.05		.1089	1268.00		.1085	1257.35	
d		.498			.530			.545			.570			.605			.640			.685			.735			.790			.885
620	1339.06		.0656	1332.50		.0690	1325.60		.0730	1318.30		.0760	1310.70		.0800	1302.70		.0840	1294.30		.0890	1285.40		.0930	1275.90		.1000	1265.90	
d		.498			.515			.540			.565			.595			.625			.660			.700			.750			.800
630	1344.04		.0639	1337.45		.0665	1331.00		.0705	1323.95		.0730	1316.65		.0770	1308.95		.0805	1300.90		.0850	1292.40		.0900	1283.40		.0950	1273.90	
d		.498			.515			.540			.565			.595			.625			.660			.700			.750			.800
640	1349.02		.0622	1342.80		.0640	1336.40		.0680	1329.40		.0700	1322.40		.0740	1315.20		.0770	1307.50		.0810	1299.40		.0850	1290.90		.0900	1281.90	
d		.498			.515			.535			.560			.580			.610			.640			.675			.720			.795
650	1354.00		.0605	1347.95		.0620	1341.75		.0655	1335.20		.0680	1328.40		.0720	1321.30		.0740	1313.90		.0775	1306.15		.0815	1298.00		.0855	1289.45	
d		.498			.515			.535			.560			.580			.610			.640			.675			.720			.795
660	1358.98		.0588	1353.10		.0608	1347.10		.0630	1340.80		.0660	1334.20		.0680	1327.40		.0710	1320.30		.0760	1312.90		.0780	1305.10		.0810	1297.00	
d		.498			.515			.530			.550			.575			.595			.625			.650			.695			.780
670	1363.96		.0572	1358.25		.0585	1352.40		.0610	1346.30		.0635	1339.95		.0660	1333.33		.0680	1326.55		.0715	1319.40		.0745	1312.25		.0775	1304.20	
d		.498			.515			.530			.550			.575			.595			.625			.650			.695			.780
680	1368.94		.0554	1363.40		.0570	1357.70		.0590	1351.80		.0610	1345.70		.0640	1339.30		.0650	1332.80		.0690	1325.90		.0710	1318.80		.0740	1311.40	
d		.498			.510			.530			.545			.565			.590			.610			.635			.665			.695
690	1373.92		.0542	1368.50		.0550	1363.00		.0575	1357.25		.0590	1351.35		.0615	1345.20		.0630	1338.90		.0665	1332.25		.0680	1325.45		.0710	1318.35	
d		.498			.510			.530			.545			.565			.590			.610			.635			.665			.695
700	1378.90		.0530	1373.60		.0530	1368.30		.0560	1362.70		.0570	1357.00		.0590	1351.10		.0610	1345.00		.0660	1338.60		.0650	1332.10		.0680	1325.30	
d		.498			.512			.525			.545			.560			.575			.595			.620			.645			.670
710	1383.88		.0516	1378.72		.0517	1373.55		.0540	1368.15		.0555	1362.60		.0575	1356.85		.0590	1350.95		.0615	1344.80		.0625	1338.55		.0655	1332.00	
d		.498			.512			.525			.545			.560			.575			.595			.620			.645			.670
720	1388.86		.0502	1383.84		.0504	1378.80		.0520	1373.40		.0540	1368.20		.0560	1362.60		.0570	1356.90		.0590	1351.00		.0600	1345.00		.0630	1338.70	
d		.498			.512			.525			.535			.555			.570			.590			.610			.635			.690
730	1393.84		.0488	1388.96		.0492	1384.05		.0510	1378.95		.0520	1373.75		.0545	1368.30		.0550	1362.80		.0570	1357.10		.0585	1351.25		.0605	1345.20	
d		.498			.512			.525			.535			.555			.570			.590			.610			.625			.690
740	1398.82		.0474	1394.08		.0478	1389.30		.0500	1384.30		.0500	1379.30		.0530	1374.00		.0530	1368.70		.0530	1363.20		.0570	1357.50		.0580	1351.70	
d		.498			.512			.525			.540			.550			.565			.580			.600			.620			.639
750	1403.80		.0460	1399.20		.0465	1394.55		.0485	1389.70		.0490	1384.80		.0515	1379.65		.0515	1374.50		.0530	1369.20		.0550	1363.70		.0565	1358.05	
d		.502			.512			.525			.540			.550			.565			.580			.600			.620			.639
760	1408.82		.0450	1404.32		.0452	1399.80		.0470	1395.10		.0480	1390.30		.0500	1385.30		.0500	1380.30		.0510	1375.20		.0530	1369.90		.0550	1364.40	
d		.502			.512			.520			.535			.545			.560			.575			.590			.605			.625
770	1413.84		.0440	1409.44		.0444	1405.00		.0455	1400.45		.0470	1395.75		.0485	1390.90		.0485	1386.05		.0495	1381.10		.0515	1375.95		.0530	1370.65	
d		.502			.512			.520			.535			.545			.560			.575			.590			.605			.625
780	1418.86		.0430	1414.56		.0436	1410.20		.0440	1405.80		.0460	1401.20		.0470	1396.30		.0470	1391.40		.0480	1387.00		.0500	1382.00		.0510	1376.90	
d		.502			.512			.520			.530			.545			.560			.570			.580			.595			.625
790	1423.88		.0420	1419.68		.0428	1415.40		.0430	1411.10		.0445	1406.65		.0455	1402.10		.0460	1397.50		.0470	1392.80		.0485	1387.95		.0490	1382.85	
d		.502			.512			.520			.530			.545			.560			.570			.580			.595			.625
800	1428.90		.0410	1424.80		.0420	1420.60		.0420	1416.40		.0430	1412.10		.0440	1407.70		.0450	1403.20		.0460	1398.60		.0470	1393.90		.0470	1389.20	
d		.504			.512			.522			.530			.540			.550			.562			.575			.590			.600
810	1433.94		.0402	1429.92		.0418	1425.82		.0412	1421.70		.0420	1417.30		.0430	1413.20		.0438	1408.88		.0447	1404.35		.0455	1399.80		.0460	1395.20	
d		.504			.512			.522			.530			.540			.550			.562			.575			.590			.600
820	1438.98		.0394	1435.04		.0408	1431.04		.0404	1427.00		.0410	1422.90		.0420	1418.70		.0426	1414.44		.0434	1410.10		.0440	1405.70		.0450	1401.20	
d		.504			.512			.522			.530			.540			.550			.562			.575			.589			.599
830	1444.02		.0376	1440.26		.0400	1436.26		.0396	1432.30		.0400	1428.30		.0410	1424.20		.0424	1420.04		.0422	1415.85		.0430	1411.55		.0440	1407.15	
d		.504			.512			.522			.530			.540			.550			.562			.575			.589			.599
840	1449.06		.0368	1445.38		.0390	1441.48		.0388	1437.60		.0390	1433.70		.0400	1429.70		.0402	1425.68		.0408	1421.60		.0420	1417.40		.0430	1413.10	
d		.504			.512			.522			.530			.540			.550			.562			.575			.589			.599
850	1454.10		.0370	1450.40		.0370	1446.70		.0380	1442.90		.0380	1439.10		.0390	1435.20		.0390	1431.30		.0403	1427.25		.0410	1423.15		.0425	1418.90	
d		.508			.516			.522			.530			.538			.546			.554			.565			.575			.589	
860	1459.18		.0362	1455.56		.0364	1451.92		.0378	1448.20		.0378	1444.40		.0388	1440.46		.0382	1436.84		.0394	1432.70		.0400	1428.90		.0400	1424.70	
d		.508			.516			.522			.530			.538			.546			.554			.565			.575			.589	
870	1464.26		.0354	1460.72		.0358	1457.14		.0344	1453.50		.0364	1449.80		.0374	1446.12		.0374	1442.38		.0383	1438.55		.0390	1434.65		.0390	1430.75	
d		.508			.516			.522			.530			.538			.546			.554			.565			.575			.589	
880	1469.34		.0346	1465.88		.0352	1462.36		.0336	1458.80		.0356	1455.24		.0366	1451.58		.0366	1447.92		.0372	1444.20		.0380	1440.40		.0380	1436.60	
d		.508			.516			.522			.530			.538			.546			.554			.565			.575			.589	
890	1474.42		.0338	1471.04		.0346	1467.38		.0348	1464.10		.0348	1460.42		.0358	1457.04		.0358	1453.46		.0366	1449.69		.0370	1446.10		.0370	1442.60	
d		.508			.516			.522			.530			.538			.546			.554			.565			.575			.589	
900	1479.50		.0330	1476.20		.0340	1472.80		.0340	1469.40		.0340	1466.00		.0350	1462.50		.0360	1459.00		.0360	1455.40		.0360	1451.80		.0360	1448.20	

considered to be significant and will agree to within plus or minus one-tenth of a Btu./lb. The following paragraph illustrates the method of using the table.

To find the enthalpy of superheated steam at a pressure of 624 pounds per square inch (absolute) and a temperature of 843 degrees (F.): From the column headed 600 pounds per square inch (absolute) and opposite the temperature line 840 degrees (F.) a base value for enthalpy, 1429.70 Btu./lb. is obtained. To correct this base value for temperature, the increment of enthalpy is obtained from the "d" line between the 840 degrees (F.) and 850 degrees (F.) lines in the 600 pounds per square inch (absolute column, this increment, 0.550 Btu./lb./° F., is then multiplied by 3, or the difference between 840 degrees (F.) and 843 degrees (F.), making the temperature correction 1.65 Btu./lb. The correction for pressure is obtained in the same manner: Referring to the "d" column between the 600 pounds per square inch (absolute) and the 700 pounds per square inch (absolute) columns for the pressure increment in the 840 degrees (F.) line, this item is found to be 0.0402 Btu./lb./psi. This pressure increment is then multiplied by 24, or the difference between 600 pounds per square inch, and 624 pounds per square inch, making the pressure correction 0.96 Btu./lb. This pressure correction is negative as the enthalpy decreases with pressure increase. The final value of enthalpy for the required condition is, therefore, 1429.70 + 1.65 — 0.96, or, 1430.39 Btu./lb.

For simplicity of solution, the work may be arranged as follows:

"Base" value 840° F., 600 psi (abs.)	1429.70 Btu./lb.
"Temperature Correction" (843-840) × 0.550 =	1.65 Btu./lb.
"Pressure Correction" (624-600) × 0.0402 =	—0.96 Btu./lb.
"Final" value: 843° F., 624 psi (abs.)	1430.39 Btu./lb.

The values of enthalpy are based upon the 1936 steam tables of Joseph H. Keenan and Fredrick G. Keyes, published by John Wiley and Sons, Inc., by permission.

THE HULL AND ITS SCREW PROPELLER.

By E. A. Stevens, Jr., Member.

Since the publication of Admiral C. W. Dyson's last book, "Screw Propellers", data has been obtained which has shown the necessity of revising some of the charts and adding new ones in order to make allowances for certain features which were not taken into consideration.

Some of the charts, given herein, have been redrawn and appear quite different from those given by Admiral Dyson, but will in many cases give practically the same results. Although some of the methods given herein might seem new, this work is not entirely original, but is a development, and in some cases a simplification, of the Dyson method of propeller analysis and design.

Part I—Power.

Effective Horsepower.

In making estimates for the shaft horsepower required for propelling a vessel at a desired speed, it is necessary to know the effective horsepower. This subject has been very carefully discussed in "Power and Speed of Ships" by the late Rear Admiral D. W. Taylor (C.C.), U.S.N., and the reader is referred to that book for making such estimates. Unfortunately, this work does not give charts for obtaining the residual resistance for speed ratios under .60. To cover this range Chart 1A has been prepared. The effective horsepower as obtained from this chart has been compared with tank results of a number of slow cargo ships having block coefficients from .77 to .80 and has been found to be quite accurate for speed ratios up to .57. Above this ratio it gives somewhat higher values than the Taylor standard series. It does, however, give results very close to those for Baker's model 56C used in a series of tests described in Vol. 38 of the Transactions of the Society of Naval Architects and Marine Engineers ("Test

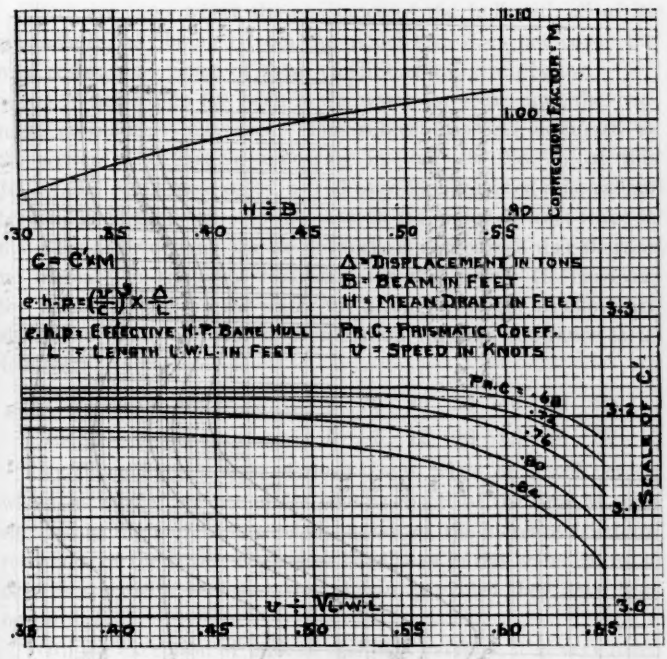


CHART 1A.

Results of a Series of Fifteen Models" by Mr. James L. Ackerson). The effective horsepower obtained from Taylor's standard series or from Chart 1A does not, however, include the power required to overcome the resistance of the appendages. It is, therefore, necessary (as in the case when a model has been tested without the appendages) to make an allowance for such.

Chart 1B gives the percentages that should be added to the bare hull effective horsepower to allow for such appendages. It must be borne in mind, however, that these curves are only approximate, and that the percentage appendage resistance for any vessel will vary with the displacement, and the only method of obtaining these resistances with accuracy is to tow the model with its appendages at the displacement for which guarantees of performance are required.

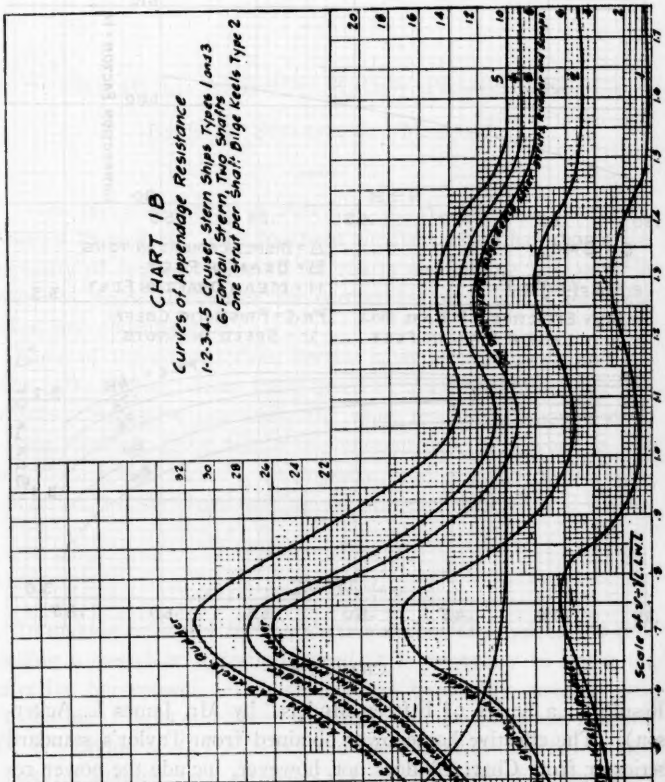


CHART 1B
Curves of Appendage Resistance
1-2-3-4-5 Cruiser Stern Ships Types 1 and 3
6-Fantail Stern, Two Shafts
One Strut per Shaft. Bilge keels Type 2
Scale of Appendage Resistance in percent of Bare Hull Resistance
Scale of V÷√L

There are cases where the resistance of the hull with all of its appendages, as obtained from the model test, does not cover the total resistance. Some of these conditions are given below.

(*a*) When a ship has been in the water for some time, the frictional resistance is increased. This subject has been discussed by the writer in an article appearing in Vol. 49, No. 4. of this JOURNAL. As in most cases, a trial over the measured mile cannot be conducted until the ship has been in the water at least three to five days after coming out of drydock, it is assumed that there is no increase in frictional resistance for this time. If, however, the vessel has been in the water for a longer period, an allowance should be made. A fair approximation for this allowance is as follows:

Days in the water.....	5	10	20	30	40	50	60
Increase in per cent.....	0	5	12	18	23	28	32

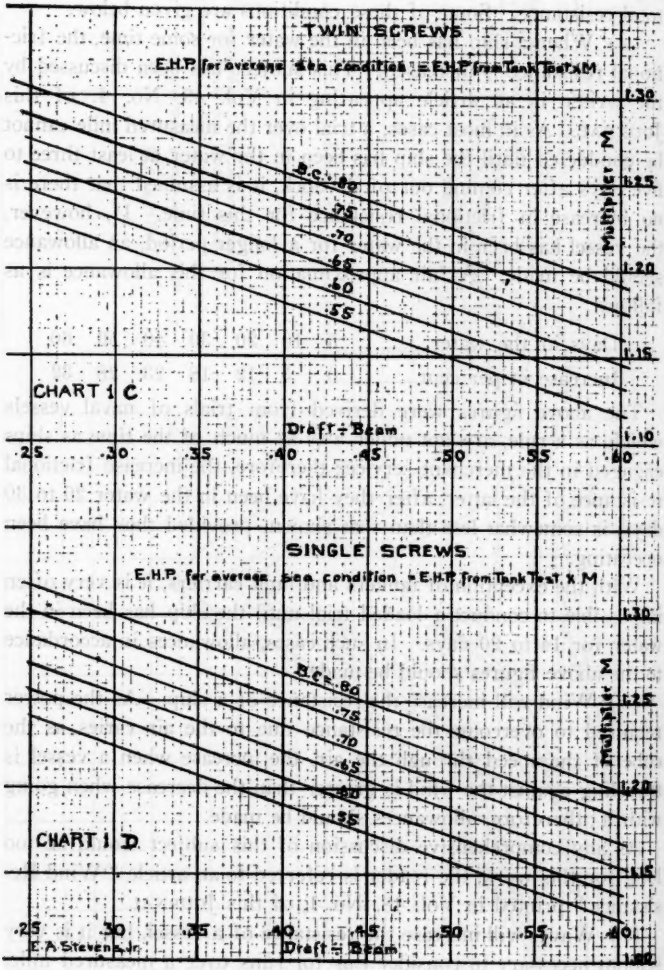
The above figures were derived from trials of naval vessels which, as a rule, are not under way as much of the time as ships engaged in the merchant service; therefore, the increase frictional resistance of the latter, after they have been in the water 20 to 30 days, is somewhat less than that shown, provided they have been operating.

With the exception of tankers and bulk carriers, it is very often impossible to conduct a loaded trial until the ship has been in the water for 14 to 20 days. In such cases, allowances in accordance to the above figures should be made.

(*b*) Wind will increase the resistance of a ship. As the power required to overcome the resistance due to the air varies as the cube of the speed through the air, the increase when a vessel is traveling against the wind is greater than the decrease when going with it; therefore, allowances should be made.

As any comprehensive discussion of this subject would be too long for this work, the reader is referred to an article "Wind Resistance" printed in Vol. 48, No. 1, of this JOURNAL.

(*c*) Waves will increase the resistance of a vessel, but it is very seldom necessary to consider this for runs over a measured mile. In service, however, it is of importance. There is no method to determine the average increase resistance due to waves, as this


TWIN SCREWS
E.H.P. for average Sea condition = E.H.P. from Tank Test x M
B.C. .80
.75
.70
.65
.60
.55
Multiplier M
1.30
1.25
1.20
1.15
1.10
CHART 1 C
Draft ÷ Beam
.25
.30
.35
.40
.45
.50
.55
.60
SINGLE SCREWS
E.H.P. for average Sea condition = E.H.P. from Tank Test x M
B.C. .80
.75
.70
.65
.60
.55
Multiplier M
1.30
1.25
1.20
1.15
CHART 1 D
.25
.30
.35
.40
.45
.50
.55
.60
E. A. Stevens, Jr.
Draft ÷ Beam

varies considerably with both the height and length of the waves and the size and shape of the vessels; however, it can be stated that ships of fine form are, as a general rule, affected adversely less than those of fuller form.

Service Factor.—In designing a propeller, the effective horsepower to be used should not be that for the measured mile under ideal conditions (freshly painted bottom, smooth sea and no wind), but that for service conditions, which include the increase resistances due to wind, waves and the condition of the ship's bottom. In order to obtain an estimate of the effective horsepower in service, the tank effective horsepower (with all appendages) is multiplied by what is known as the Service Factor, approximate values of which can be obtained from Charts 1C and 1D. These charts were made for ships of 400 feet waterline operating in the North Atlantic. If the vessel is of a different length, apply the following corrections:

Ships longer than 400 feet L.W.L., subtract .01 for every 100 feet above 400.

Ships under 400 feet L.W.L., add .01 for every 100 feet under 400.

If the vessel is operating in other trades than the North Atlantic, the service factor may not be the same. In such cases, the designer should use his judgment.

ENGINE POWER.

It will be necessary to convert indicated horsepower to shaft horsepower as the latter will be used in all cases as the engine power in this work. In order to make this conversion, it is essential that the mechanical efficiency of the engine be known. As this varies with different engines, the values given should be considered as only approximate.

Reciprocating Steam Engine.—The mechanical efficiencies of large marine engines using mean referred pressures as a basis are given below:

M.R.P.	20	25	30	35	40	45	50
Mech. Eff.....	.886	.906	.919	.928	.935	.939	.941

The above are the mean values obtained from tests of two large quadruple expansion engines with no attached auxiliaries, such as air, bilge, sanitary and feed pumps. If attached auxiliaries are used, the mechanical efficiency will be somewhat less. If forced lubrication is used, multiply the above values by 1.03.

If the low pressure cylinder is under 70 inches diameter (or 50 inches if there are two low pressure cylinders), the mechanical efficiency obtained above should be corrected by multiplying by the following correction factor (M):

Diam. L.P. Cylinder.....	30	40	50	60	70
M	.920	.955	.978	.992	1.00

At fractional loads the mechanical efficiency, as a rule, will decrease, although there are cases where propeller analyses indicate that this decrease does not seem to occur with large engines fitted with forced lubrication.

Diesel Engines.—As the mechanical efficiency of Diesel Engines vary considerably, depending upon the size and type of engine and the attached auxiliaries, this data should, when possible, be obtained from the manufacturer. If, however, it is not possible to obtain this data, the following may be taken as a fair average for engines having cylinders of 25 inches diameter and over when there are no attached auxiliaries.

Solid injection engines or air injection engines with independent driven air compressors.

	Mech. Eff.	
Type of Engine	*Cross-head*	*Trunk Piston*
4 cycle	.82	.85
2 cycle—scavenging pump attached..	.80	.82
2 cycle—scavenging pump independent	.86	.88

Air injection engines with air compressor attached to engine.

	Mech. Eff.	
Type of Engine	*Cross-head*	*Trunk Piston*
4 cycle	.75	.76
2 cycle—scavenging pump attached..	.73	.74
2 cycle—scavenging pump independent	.79	.80

Should the cylinders be less than 25 inches diameter, the above mechanical efficiencies should be corrected by multiplying the above values by the following correction factors:

Diameter of cylinder, inches	20	15	10
Correction factor	.985	.96	.92

The above mechanical efficiencies are for engines operating with indicated mean effective pressures of about 90 pounds per square inch for four cycle and 80 pounds per square inch for two cycle engines. If the indicated mean effective pressures are less than the above figures and/or any pumps, other than the fuel injection pump, are attached to the engine, the mechanical efficiency will be somewhat less.

Part 2—The Hull.

In his last book Admiral Dyson states the following: "The problem as it stands is not the 'Problem of the Screw Propeller' it is the 'Problem of the Ship's Hull with its Screw Propeller' and of the two elements entering into it, the first, the hull, is by far the more difficult to handle in fixing with any degree of accuracy the set of conditions required."

Admiral D. W. Taylor has stated on several occasions that the hull has a decided effect on the propeller's performance and often more can be gained by changing the ship's lines than by any improvement in the propeller.

The above statements coming from such noted authorities should leave little doubt of the desirability of having some method of obtaining the effect of the hull on the operation of the propeller.

HULL DATA REQUIRED.

Before the effect of the hull on the propeller can be determined, it is necessary to have the following data:

L = Length in feet. On twin screw ships this is equal to the length of the water line (L.W.L.). On single screw ships L. is equal to the length between perpendiculars (L.B.P.).

B = Beam in feet. This is the greatest immersed beam. On steel ships it is taken as the molded beam, but on wooden and composite ships, or steel vessels sheathed with wood, the beam should be measured to the outside of the planking.

H = Mean draft in feet. On steel ships with plate keels this is taken as the molded mean draft. On ships where the keel extends below the hull, as with wooden or composite ships, or steel ships fitted with a bar keel, the mean draft is measured to the bottom of the hull excluding the keel.

Δ = Displacement in Tons (2240 lbs.) at draft H. The molded displacement should be used for steel ships. If, however, the mean draft has been taken to the bottom of the plate keel, the displacement should include the plating; this will give approximately the same block co-efficient as if the molded draft and displacement were used. On wooden and composite ships the total displacement, including planking, should be used.

Area M.S. = Area of the submerged Midship Section in square feet.

Area Sta. 19 = Area in square feet of a station at a distance of 5 per cent of L forward of the rudder post. This item is necessary only for single screw ships of Type 2. The shape of this section should be noted as this as well as its area has an influence on the propulsive coefficient. If the shape of this section is hollow, as shown by the full line in the figure on Chart 4E, it may be called Y shape, but should it be straight or nearly so, as shown by the dotted line, it is V shaped.

L.A.B. = Length of after body in feet. This is the length of what may be called the effective after body. It is obtained as follows: As shown on Figure 1A, draw a water line through the middle of the turn of the bilge (W.L. No. 1) and at a distance from the centerline equal to $\frac{3}{8}$ of the beam draw a buttock line. Note the point (A) where the water line begins to turn in and the point (B) where the above buttock line begins to rise. A mean between these two points is the forward point of measurement. The after point of measurement is the same as L. Should, however, the above water line turn in sharply as shown

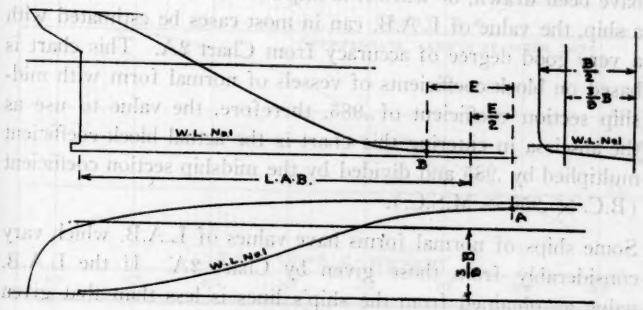


Fig. 1A

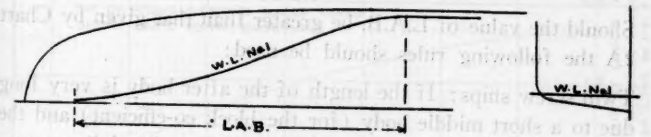


Fig. 1B

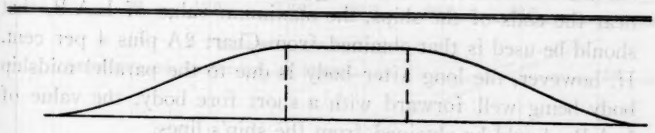


Fig. 2A

Fig. 2B

on Figure 1B, the full length of the actual after body should be taken. For preliminary work before the lines of the ship have been drawn, or when it is impossible to obtain the lines of a ship, the value of L.A.B. can in most cases be estimated with a very good degree of accuracy from Chart 2A. This chart is based on block coefficients of vessels of normal form with midship section coefficient of .985, therefore, the value to use as the abscissa in entering this chart is the actual block coefficient multiplied by .985 and divided by the midship section coefficient (B.C. $\times$ .958 $\div$ M.S.C.).

Some ships of normal forms have values of L.A.B. which vary considerably from those given by Chart 2A. If the L.A.B. value as obtained from the ship's lines is less than that given by the use of Chart 2A, the former should be used.

Should the value of L.A.B. be greater than that given by Chart 2A the following rules should be used:

Twin screw ships: If the length of the after body is very long due to a short middle body (for the block co-efficient) and the curve of sectional areas turn in gradually, with full sections near the ends of the ships, the maximum value of L.A.B. that should be used is that obtained from Chart 2A plus 4 per cent. If, however, the long after body is due to the parallel midship body being well forward with a short fore body, the value of L.A.B. should be obtained from the ship's lines.

Single Screw Ships: If the L.A.B. obtained from the ship's lines is considerably greater than that given by Chart 2A, use a value that is 4 per cent greater than that given by this chart.

H′ = This value is only used in connection with L.A.B. for determining the effect of the hull on the revolutions. For twin screw ships of normal form, it is equal to H. If the ship is fitted with a transom stern, the lower part of which is submerged, H′ is the distance in feet from the base line to the bottom of the transom. For single screw ships of normal form it is the distance of the upper tip of the propeller above the base line. Chart 2B has been prepared so that this value (H′) for single screw ships may be estimated when the lines of the

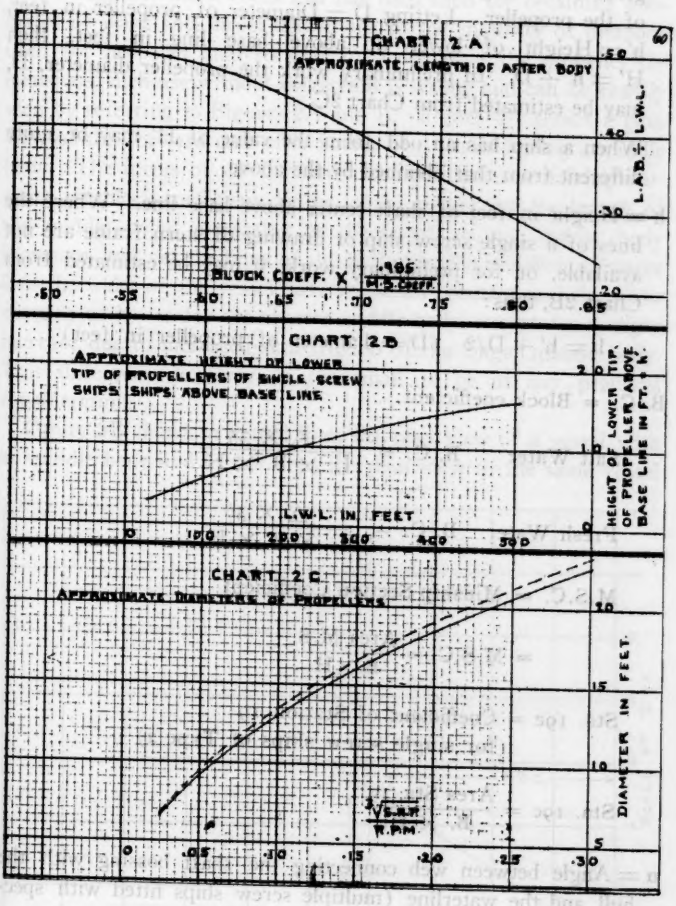


CHART 2 A
APPROXIMATE LENGTH OF AFTER BODY
L.A.B. ÷ L.W.L.
BLOCK COEFF. X .965 / M-S COEFF.
.50 .55 .60 .65 .70 .75 .80 .85
.50 .40 .30 .20
CHART 2 B
APPROXIMATE HEIGHT OF LOWER TIP OF PROPELLERS OF SINGLE SCREW SHIPS ABOVE BASE LINE
HEIGHT OF LOWER TIP OF PROPELLER ABOVE BASE LINE IN FT. = H.
L.W.L. IN FEET
0 100 200 300 400 500
2.0 1.0 0
CHART 2 C
APPROXIMATE DIAMETERS OF PROPELLERS
DIAMETER IN FEET.
√S.H.P. / R.P.M.
0 .05 .10 .15 .20 .25 .30
20 15 10 5

ship or the drawing of the stern frame are not available. It gives the average distance above the base line of the lower tip of the propeller. Letting D = Diameter of propeller in feet, h′ = Height of lower tip above base line in feet, then H′ = h′ + D. In preliminary work the propeller diameter, D, may be estimated from Chart 2C.

When a ship has an odd form the value of H′ may be quite different from that obtained by the above.

h = Height in feet of shaft center above base line. Where the lines of a single screw ship or drawing of stern frame are not available, or for preliminary work, h may be estimated from Chart 2B, thus:

$$h = h' + D/2 \quad (D = \text{diameter of propeller in feet})$$

B. C. = Block coefficient

Salt Water $\quad B.\,C. = \dfrac{\Delta \times 35}{L. \times B. \times H.}$

Fresh Water $\quad B.\,C. = \dfrac{\Delta \times 36}{L. \times B. \times H.}$

M.S.C. = Midship Section Coefficient

$$= M.S.C. = \frac{\text{Area M.S.}}{B. \times H.}$$

Sta. 19c = Coefficient of Station 19
(for single screw ships of Type 2)

$$\text{Sta. 19c} = \frac{\text{Area Sta. 19}}{B. \times H.}$$

α = Angle between web connecting the shaft bossing with the hull and the waterline (multiple screw ships fitted with spectacle frames—See Figure on Chart 4C).

T.C. = Minimum clearance between the tip of the propeller and the hull in feet (For twin and four screw ships and wing propellers of triple screw vessels).

BLOCK COEFFICIENT.

Although the block coefficient has been used for obtaining estimates of wake and thrust deductions in the majority of methods used for propeller analysis and for calculating the propeller dimensions, such use without correction is wrong and can be readily seen by referring to Figures 2A and B. Figure 2A shows a plan view of the forward and after bodies of a vessel having a middle body. The forms of the forward and after bodies remaining constant, the block coefficient can be varied by varying the length of the middle body, increasing as the length of the middle body increases, and yet the lines of the hull directing the flow of water into the wake of the vessel have remained constant and the wake remains as it was for the shorter middle body, except for a slight increase due to increased skin friction of the longer middle body. This difference, however, is too small to be of any practical consideration.

Figure 2B shows in dotted lines the plan view of a vessel with no middle body and in full lines the plan view of the same vessel

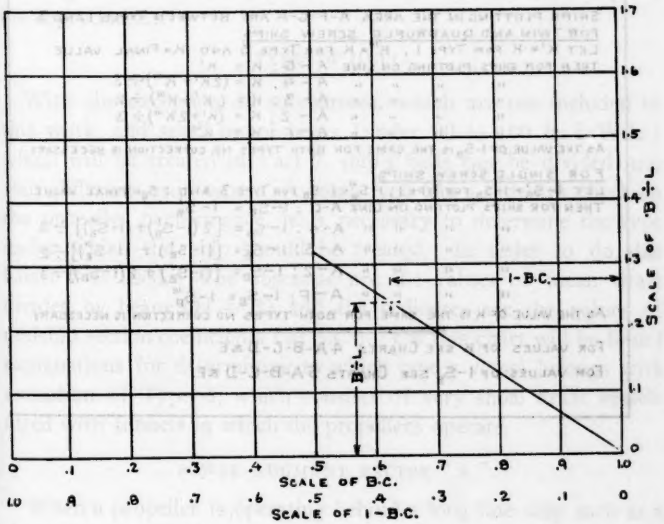


FIGURE 3.

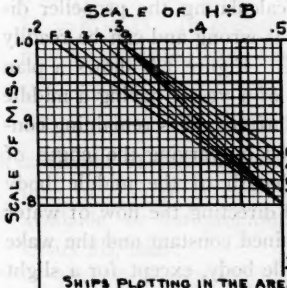


CHART 3

Ships plotting on Line G-A-B are Type 1

Ships plotting to the right of Line G-H-A-B are Type 2.

Ships plotting to the left of Line F-A-B are Type 3

Ships plotting in the area A-G-H-A are between Types 1 and 2.

For multiple and single screw ships

Let $K' = K$ for Type 1, $K'' = K$ for Type 2, and K = Final value

Then for ships plotting on line

G-A ; $K = K'$

G-7 ; $K = (2K' + K'') \div 3$

G-8 ; $K = (K' + K'') \div 2$

G-9 ; $K = (K' + 2K'') \div 3$

G-H ; $K = K''$

As the value of $1-S_B$ is the same for both types no correction is necessary

Ships plotting in the area A-F-G-A are between Types 1 and 3

For Twin and Quadruple screw ships

Let $K' = K$ for Type 1, $K''' = K$ for Type 3 and K = Final value

Then for ships plotting on line

A-G ; $K = K'$

A-4 ; $K = (2K' + K''') \div 3$

A-3 ; $K = (K' + K''') \div 2$

A-2 ; $K = (K' + 2K''') \div 3$

A-F ; $K = K'''$

As the value of $1-S_B$ is the same for both types no correction is necessary

For single screw ships

Let $1-S_B = 1-S_B$ for Type 1, $1-S_B''' = 1-S_B$ for Type 3, and $1-S_B$ = Final value

Then for ships plotting on line

A-G ; $1-S_B = 1-S_B'$

A-4 ; $1-S_B = [2(1-S_B') + (1-S_B''')] \div 3$

A-3 ; $1-S_B = [(1-S_B') + (1-S_B''')] \div 2$

A-2 ; $1-S_B = [(1-S_B') + 2(1-S_B''')] \div 3$

A-F ; $1-S_B = 1-S_B'''$

As the value of K is the same for both types no correction is necessary.

For values of K see Charts 4A-B-C-D & E

For values of $1-S_B$ see Charts 5A-B-C-D & E

with fore and aft lines increased in length. These two vessels will have the same block coefficient, but the wakes will be widely different.

If the block coefficient is varied by varying the length of the middle body, and plotting the various vessels on a chart using block coefficients (B.C.) as abscissae and beam divided by length (B ÷ L) as ordinates as shown on Figure 3 it will be found that these points will lie on a straight line which will intersect the lower right corner where B.C. = 1.0 and B ÷ L = 0. The value of (B ÷ L) ÷ (1-B.C.) of all points on this line will be constant and the flow of water at the stern will be the same, provided the fore and after bodies are the same in all cases. This fact has been taken advantage of in developing Charts 4A-B-D and F for determining the value of the power augment factor "K" and Charts 5A for obtaining the effect that the fullness of the hull has on the revolutions of propeller, the latter being known as the slip block coefficient (S.B.C.).

The idea of using the value of (B ÷ L) ÷ (1-B.C.) was originated by Mr. C.H.E. Meyer to whom the writer wishes to express his appreciations.

TYPES OF HULL.

With the exception of submarines, which are not included in this work, and small motor boats (under 90 to 100 ft. L.W.L.) which will be treated in Part 7, ship's hulls can be divided into four different main types. As each type has a different effect on the propeller performance, it is necessary to determine the type under which the ship should be treated. In order to do this Chart 3 is used. The abscissae are the values of mean draft divided by beam (H ÷ B) and the ordinates are the values of midship section coefficients (M.S.C.). On this chart will be found explanations for determining to which type a vessel belongs with exception of Type 4, which consists of very shoal draft vessels fitted with tunnels in which the propellers operate.

POWER AUGMENT FACTOR "K".

When a propeller is operating behind a long fine ship such as a destroyer there is little or no loss due to the influence of the hull.

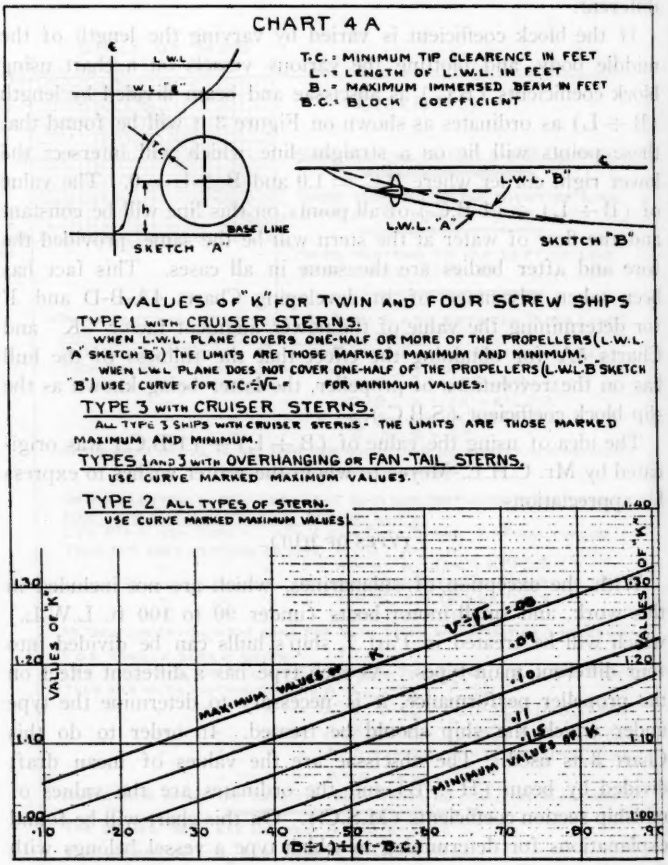


CHART 4 A
T.C. = MINIMUM TIP CLEARENCE IN FEET
L. = LENGTH OF L.W.L. IN FEET
B. = MAXIMUM IMMERSED BEAM IN FEET
B.C. = BLOCK COEFFICIENT
L.W.L.
W.L. "B"
T.C.
h
BASE LINE
SKETCH "A"
L.W.L. "B"
L.W.L. "A"
SKETCH "B"
VALUES OF "K" FOR TWIN AND FOUR SCREW SHIPS
TYPE 1 WITH CRUISER STERNS.
WHEN L.W.L. PLANE COVERS ONE-HALF OR MORE OF THE PROPELLERS (L.W.L. "A" SKETCH "B") THE LIMITS ARE THOSE MARKED MAXIMUM AND MINIMUM.
WHEN L.W.L PLANE DOES NOT COVER ONE-HALF OF THE PROPELLERS (L.W.L. "B" SKETCH "B") USE CURVE FOR T.C. ÷ √L = FOR MINIMUM VALUES.
TYPE 3 WITH CRUISER STERNS.
ALL TYPE 3 SHIPS WITH CRUISER STERNS - THE LIMITS ARE THOSE MARKED MAXIMUM AND MINIMUM.
TYPES 1 AND 3 WITH OVERHANGING OR FAN-TAIL STERNS.
USE CURVE MARKED MAXIMUM VALUES.
TYPE 2 ALL TYPES OF STERN.
USE CURVE MARKED MAXIMUM VALUES.
VALUES OF "K"
MAXIMUM VALUES OF K
MINIMUM VALUES OF K
V ÷ √L = .08
.09
.10
.11
.115
1.30
1.20
1.10
1.00
.10
.20
.30
.40
.50
.60
.70
.80
(B ÷ L) ÷ (1 - B.C.)

If, however, the hull exceeds a certain fullness (depending upon its type and dimensions) there is a loss in efficiency which results in the necessity of increasing the power in order to maintain a given speed. This increase in power is known as the "Power Augment Factor" and is designated by "K" which is obtained from Charts 4A, 4B, 4D and 4F.

Twin and Four Screw Vessels.—The value of K for vessels of Types 1 and 3 having cruiser sterns will depend upon the ratio of the minimum clearance between the tips of the propellers and the hull to the square root of the water line length as well as the value of (B ÷ L) ÷ (1-B.C.) and are obtained from Chart 4A.

Rules for obtaining the value of "K" of twin and four screw vessels are given on Chart 4A.

Correction for Spectacle Frames.—When the shafts other than those on the fore and aft center line of the vessel are carried in bosses which are connected to the hull with webs (Spectacle Frame, sometimes called Lunborg stern), the value of K will vary as the angle of the web is changed. The correction factors to be applied to the values of K obtained from Chart 4A are found in Chart 4C. The abscissae of this chart are the angles between the after end of the web and a line parallel to the water line (angle α see Figure on Chart 4C). The ordinates are the correction factors.

Let $K' =$ value of K from Chart 4A

$K =$ corrected value of K

$K.C.F._w =$ correction factor from Chart 4C

Then $K = K' \times K.C.F._w$

Single Screw Ships.—Chart 4D gives the values of K for single screw ships fitted with single plate rudders. At the top of this chart is a list giving the line to use for the various types.

As explained previously (See Area Sta. 19 and Sta. 19C under "HULL DATA REQUIRED") the shape and fullness of Station 19 of single screw vessels of Type 2 has an influence on the performance of the propeller. In order to determine which line on

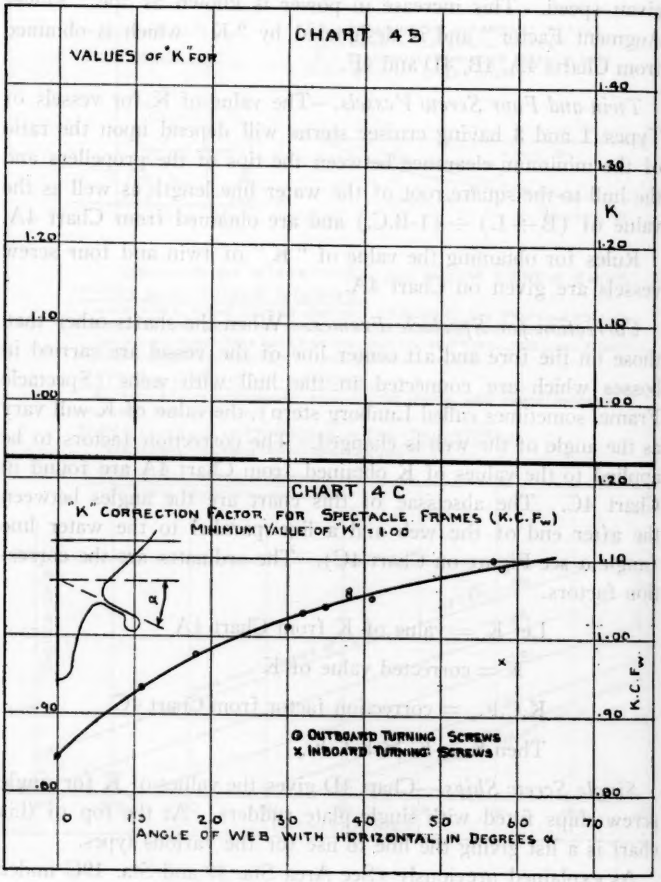


CHART 4B
VALUES OF "K" FOR
1.40
1.30
K
1.20
1.10
1.00
CHART 4C
"K" CORRECTION FACTOR FOR SPECTACLE FRAMES (K.C.F.w)
MINIMUM VALUE OF "K" = 1.00
α
1.20
1.10
1.00
.90
.80
K.C.F.w
O OUTBOARD TURNING SCREWS
X INBOARD TURNING SCREWS
0
10
20
30
40
50
60
70
ANGLE OF WEB WITH HORIZONTAL IN DEGREES α

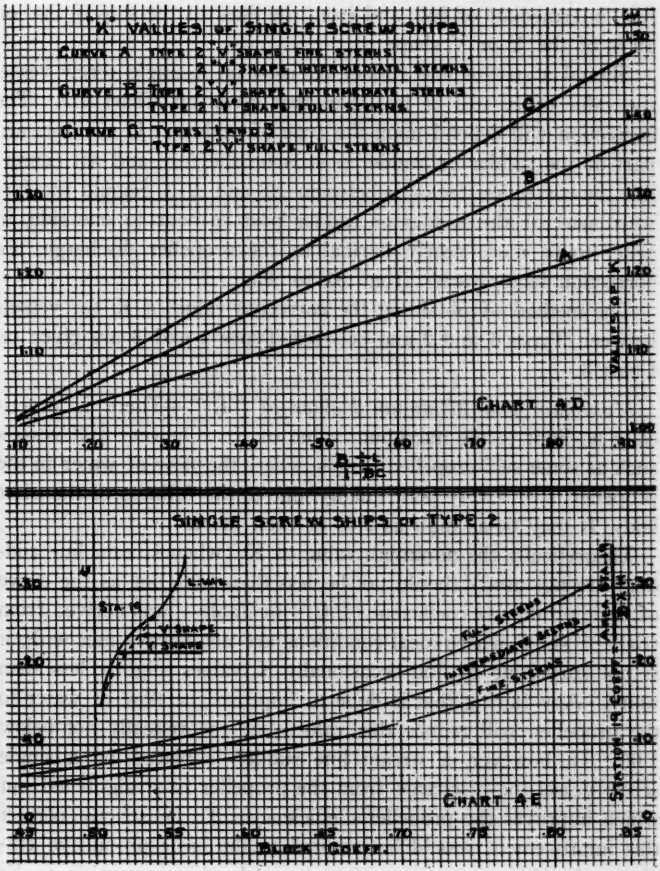

"K" VALUES OF SINGLE SCREW SHIPS.
CURVE A TYPE 2 "V" SHAPE FINE STERNS
2 "U" SHAPE, INTERMEDIATE STERNS
CURVE B TYPE 2 "V" SHAPE, INTERMEDIATE STERNS
TYPE 2 "U" SHAPE FULL STERNS
CURVE C TYPES 1 AND 3
TYPE 2 "U" SHAPE FULL STERNS
VALUES OF K
CHART 4D
SINGLE SCREW SHIPS OF TYPE 2
CHART 4E
SLIDING OF SCREW - ANGLE OF STERN
FINE STERNS
INTERMEDIATE STERNS
FULL STERNS

Chart 4D to use for obtaining K, Chart 4E has been prepared, the abscissae being block coefficients and the ordinates Station 19 coefficients. From this chart it can be determined if the stern is fine, intermediate or full, and from the lines of the ship if Station 19 is of the Y or V shape.

Should a stream line rudder be used on a single screw vessel the value of K will be less than that given on Chart 4D. The correction will depend upon the type of rudder. If the forward part of the rudder or rudder post is on the fore and aft centerline the correction factor is between .88 and .91. Should the forward part of the rudder or rudder post be bent to one side, in order to be more nearly normal to the propeller discharge, the correction factor varies from .81 to .85.

The method of obtaining the corrected value of K for vessels between Types 1 and 2 and between Types 1 and 3 are given on Chart 3.

Maximum and Minimum Values of "K".—There is practically no limit to the maximum value of K. On very full vessels such as some self-propelled barges, where the flow of water to the propeller is of a very poor quality, due to the fullness of the after body, K may be as high as 2.00 or even more.

Theoretically the minimum value of K (when stream line rudders are not used) should not be less than 1.00, but as the curve of basic propulsive coefficient (P.C.') in Table 1 was derived from trials of ships, where ideal conditions are not always met with, it is not surprising to find occasionally K values slightly less than unity. Any fairing that is done to improve the flow of water to the propeller will reduce this value, but theoretically not below 1.00. If, however, a stream line rudder is placed on a single screw vessel the discharge from the propeller is straightened out which improves the propeller's efficiency. To be strictly correct this improvement should be credited to the basic propulsive efficiency, but it is more convenient to apply this correction to K, therefore on some very fine single screw ships with a low normal power augment factor the correction factor for stream line rudders will bring the corrected K below unity.

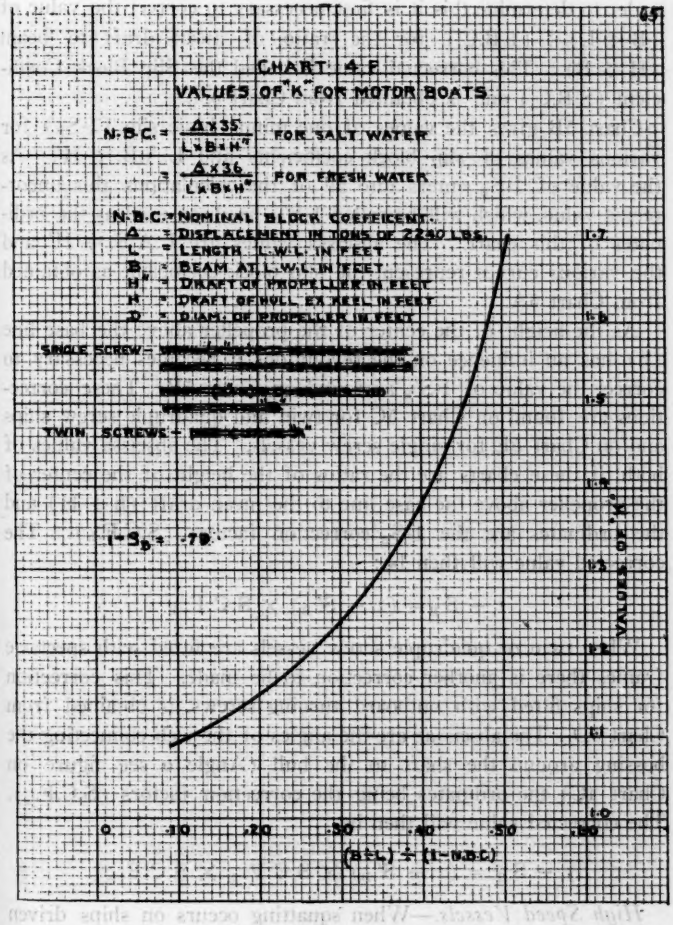


CHART 4.F
VALUES OF "K" FOR MOTOR BOATS
N.B.C. = Δ x 35 / L x B x H" FOR SALT WATER
= Δ x 36 / L x B x H" FOR FRESH WATER
N.B.C. = NOMINAL BLOCK COEFFICIENT.
Δ = DISPLACEMENT IN TONS OF 2240 LBS.
L = LENGTH L.W.L. IN FEET.
B = BEAM AT L.W.L. IN FEET.
H" = DRAFT OF PROPELLER IN FEET.
H = DRAFT OF HULL EX KEEL IN FEET.
D = DIAM. OF PROPELLER IN FEET.
SINGLE SCREW –
TWIN SCREWS –
1-S = .79
VALUES OF "K"
1·7
1·6
1·5
1·4
1·3
1·2
1·1
1·0
0
·10
·20
·30
·40
·50
·60
(B÷L) ÷ (1-N.B.C.)

BASIC SLIP "S_B".

The effect of the hull on revolution is called the Basic Slip. In order to determine this it is first necessary to correct the value of $(B \div L) \div (1\text{-B.C.})$ for the ratio of mean draft to beam $(H \div B)$. This corrected value is called the Slip Block Coefficient (S.B.C.) and is obtained from Chart 5A.

Chart 5B gives the values of one minus basic slip $(1\text{-}S_B)$ for various values of slip block coefficients and L.A.B. $\div$ H'. As the value of $1\text{-}S_B$ enters into all of the calculations this uncorrected value, which will be called $1\text{-}S'_B$, has been used as the ordinates of this chart. The abscissa values are L.A.B. $\div$ H' and the various curves represent the slip block coefficient as obtained from Chart 5A.

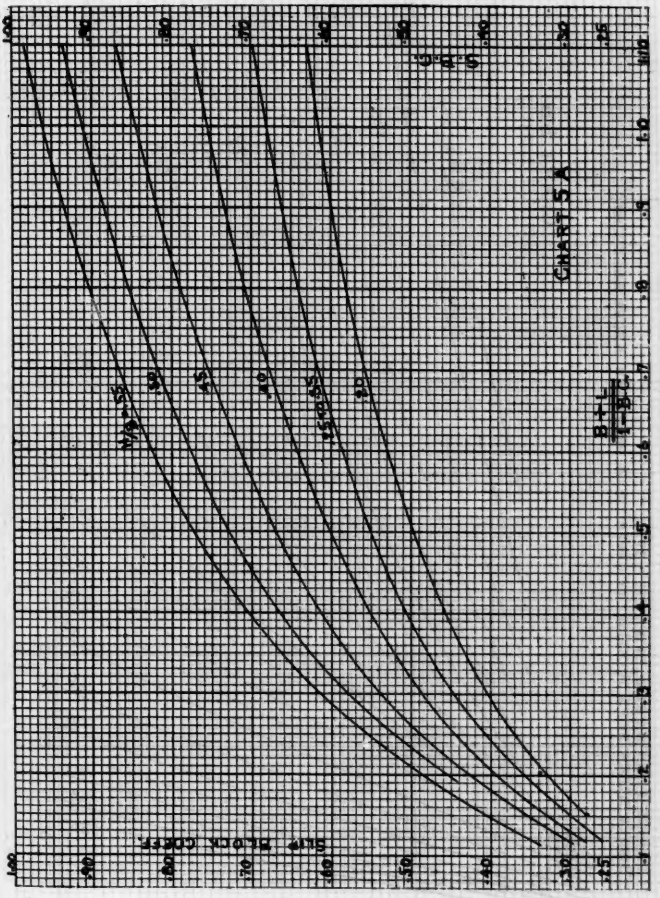
As the height of the center of the propeller above the base line (h) has an influence on the revolutions it becomes necessary to correct the value of $1\text{-}S'_B$ as found on Chart 5B. These corrections are found on Chart 5C for twin and quadruple screw ships and on Chart 5E for single screw vessels. The abscissa values of both of these charts are the ratios of the height of the center of the propeller above the base line to the mean draft $(h \div H)$ and the ordinates are the $1\text{-}S_B$ correction factors $(S.C.F._h)$. The corrected value will then be:

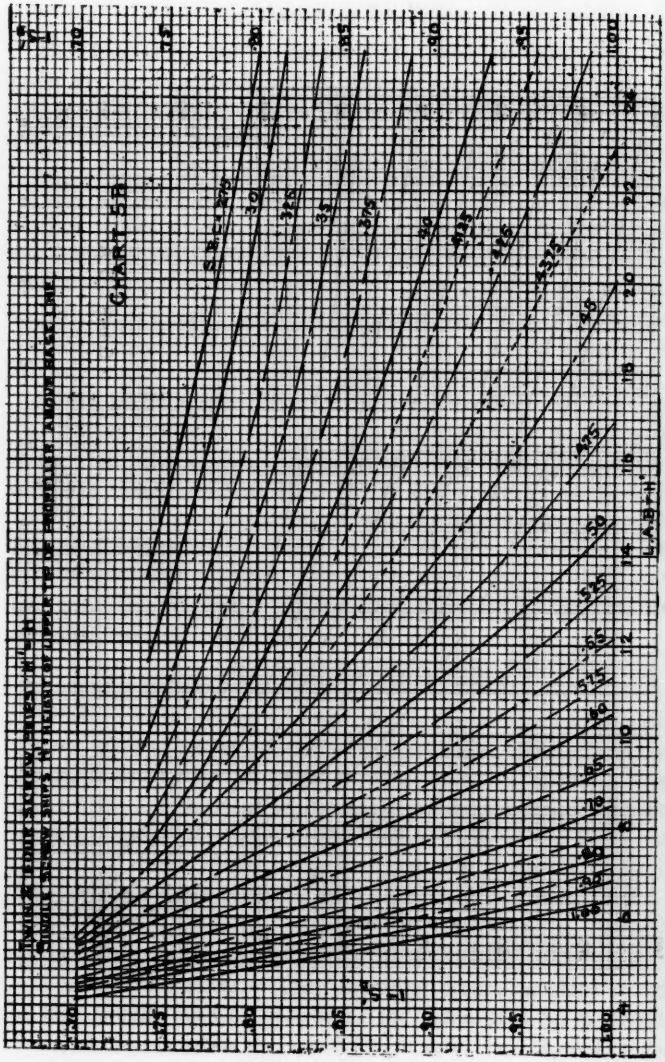
$$1 - S_B = (1 - S'_B) \times S.C.F._h$$

When twin or quadruple screw vessels are fitted with spectacle frames there is another correction to be made. This correction for ships fitted with outboard turning screws, is obtained from Chart 5D. The abscissae are the angles of the web connecting the bossing around the shaft to the hull (Angle α see figure on Chart 4C) the ordinates being the correction factors $(S.C.F._w)$. The corrected value will then be

$$1 - S_B = (1 - S'_B) \times S.C.F._h \times S.C.F._w$$

High Speed Vessels.—When squatting occurs on ships driven at high speeds such as destroyers the value of $1\text{-}S_B$ in certain cases, decreases when the speed length ratio $(v \div L.W.L.)$ exceeds a certain value. This value will depend upon a number of


Slip Block Coeff.
S.H.P.
Chart 5 M



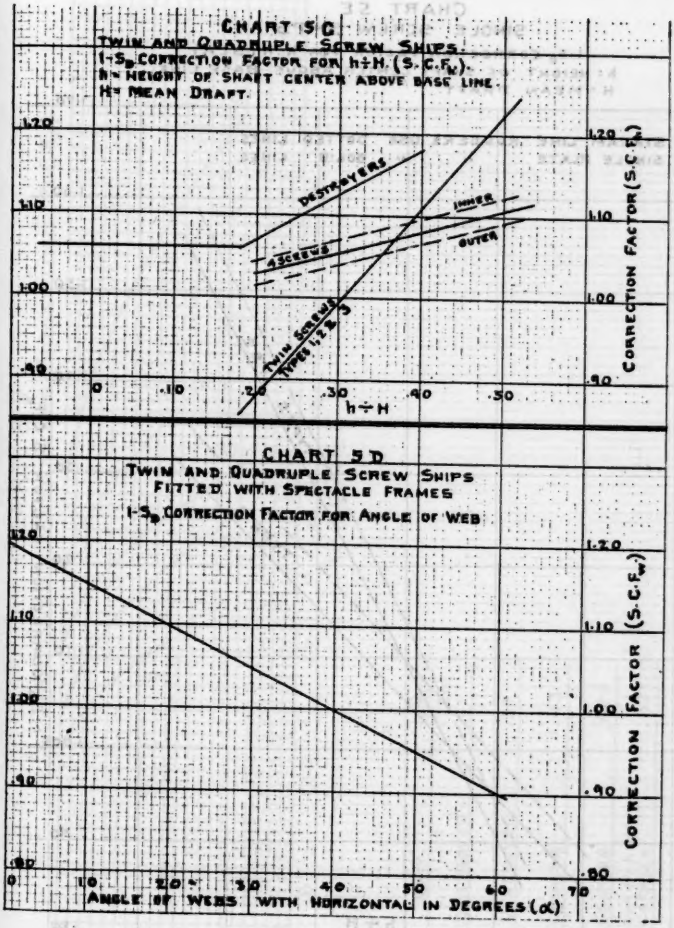


CHART 5C
TWIN AND QUADRUPLE SCREW SHIPS.
1-S_p CORRECTION FACTOR FOR h÷H. (S.C.F_w).
h= HEIGHT OF SHAFT CENTER ABOVE BASE LINE.
H= MEAN DRAFT.
1.20
1.10
1.00
.90
DESTROYERS
INNER
OUTER
4 SCREWS
TWIN SCREWS TYPES 1, 2 & 3
0
.10
.20
.30
.40
.50
h÷H
CORRECTION FACTOR (S.C.F_w)
CHART 5D
TWIN AND QUADRUPLE SCREW SHIPS
FITTED WITH SPECTACLE FRAMES
1-S_p CORRECTION FACTOR FOR ANGLE OF WEB
1.20
1.10
1.00
.90
.80
0
10
20
30
40
50
60
70
ANGLE OF WEBS WITH HORIZONTAL IN DEGREES (α)
CORRECTION FACTOR (S.C.F_w)

CHART 5E
SINGLE SCREW SHIPS

$1-S_B$ CORRECTION FACTOR FOR $h \div H$ (S.C.F.$_h$).
h = HEIGHT OF SHAFT CENTER ABOVE BASE LINE.
H = MEAN DRAFT.

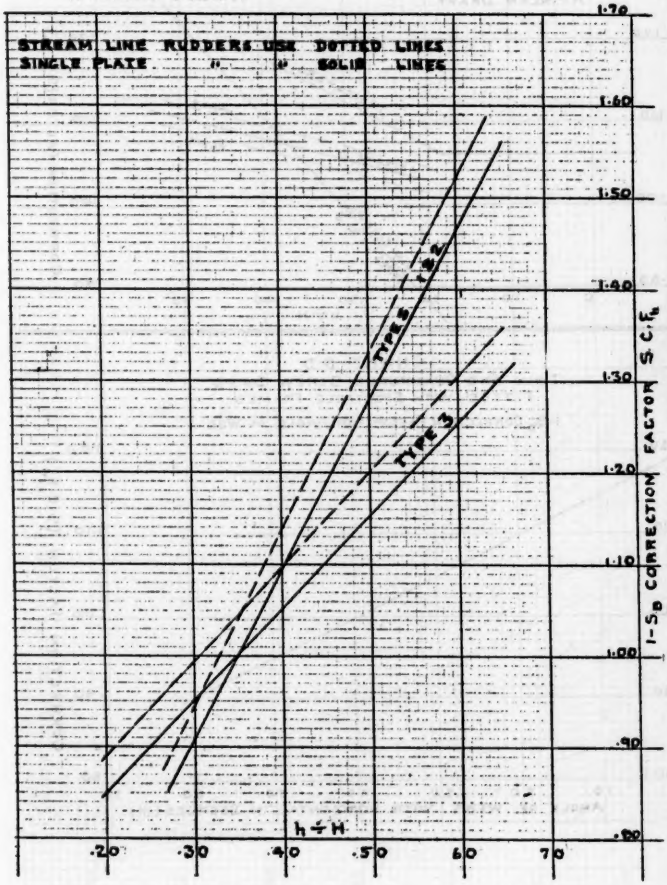
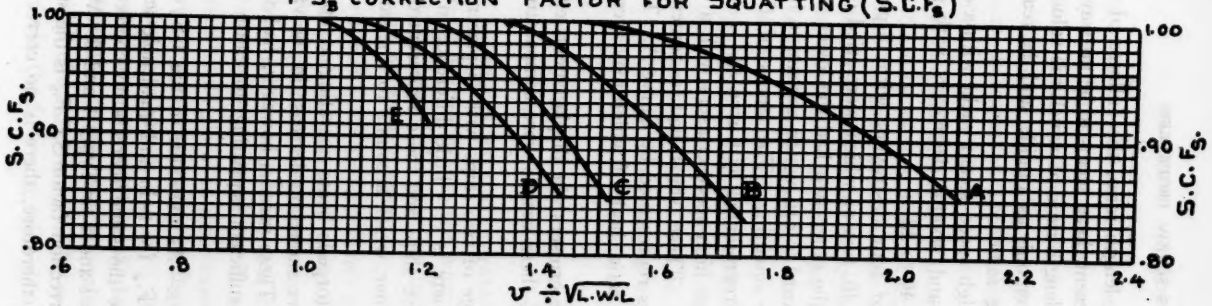


CHART 5 F

$1-S_B$ CORRECTION FACTOR FOR SQUATTING ($S.C.F_B$)



CURVE	SHP	$\frac{\Delta}{(\frac{LWL}{100})^3}$
A	DESTROYER	.41
B	DESTROYER	.39
C	CRUISER	.43
D	CRUISER	.624
E	AIR PLANE CARRIER	.631

factors such as the displacement length ratio, the fullness of the stern and the resistance. At present there seems to be no way of definitely determining the speed length ratio at which the value of $1\text{-}S_B$ is affected. Chart 5F is given only as a guide, and the reader should use his judgment in using same.

Curve A is an average of a high speed steam yacht with a displacement length ratio of 28.8 and several destroyers with displacement length ratios of from 40 to 42.

Curve B gives the average of four destroyers with displacement-length ratios from 38.5 to 40, two of which were tried over the Delaware Breakwater mile, the other two had the lower strut arm inclined upwards from forward to aft, which appears to have had the effect of increasing the squatting. The sterns of these vessels were sharper than those represented by Curve A.

Curve C is the average of two high speed cruisers with a displacement length ratio of 43. These ships had a somewhat sharper stern than the destroyers represented by Curve A. They also had higher midship section ratio, both of which may account for the earlier squatting.

Curve D is the average of five cruisers, the displacement length ratio being 62.4. The sterns of these vessels were comparatively wide.

Curve E represents the average of two ships with displacement length ratio of 63. Although there is very little difference in the displacement-length ratio of these ships and those represented by curve D, the sterns were much finer, which accounts for the earlier squatting.

In some cases destroyers or torpedo boats operating at speed-length ratios from 1.8 to 2.2 have shown no decrease in the value of $1\text{-}S_B$ at the higher speeds. These vessels, however, had very broad sterns which prevented sufficient squatting to cause any decrease in the value of $1\text{-}S_B$.

Vessels of Intermediate Types.—As the value of the $1\text{-}S_B$ correction factors $S.C.F._h$ and $S.C.F._w$ for twin and quadruple screw vessels of Types 1, 2 and 3 are the same there is no correction when a vessel is between Types 1 and 2 or Types 1 and 3. With single screw vessels the $1\text{-}S_B$ correction factor $S.C.F._h$ is the same for vessels of Types 1 and 2, therefore, there is no correction

when a vessel is between these two. When a single screw vessel is between Types 1 and 3 a correction is necessary. At the bottom of Chart 3 will be found rules for making such corrections.

It should be noted that for single screw ships of Type 1 the $1\text{-}S_B$ is the same as Type 2, but their K value is the same as Type 3.

Maximum and Minimum Values of $1\text{-}S_B$.—The maximum value of $1\text{-}S_B$ to be used in design calculations should not exceed 1.04 to 1.06. On some vessels where $1 - S'_B$ obtained from Chart 5B is high, the application of the correction factor $S.C.F._h$ may make the corrected value greater than the above figure. In such cases 1.04 to 1.06 should be used as the value of $1\text{-}S_B$ in making calculations for revolutions.

Ships having a high value of L.A.B. $\div$ H' such as torpedo boats and destroyers have a minimum value of $1\text{-}S_B$ of .84 to .86, except small motor boats (80 to 100 ft. L.W.L. and under) where this value is .78. If, however, the vessel is quite full and has a low value of L.A.B. $\div$ H' there is no limit to the minimum value of $1\text{-}S_B$.

The lower limit of the value of $1\text{-}S_B$ when this value is reduced due to squatting is about .74.

RULES FOR OBTAINING VALUES OF "K" AND "$1\text{-}S_B$".

Triple Screw Ships.—The wing propellers of triple screw ships should be treated the same as twin screws and the center wheel the same as a single screw.

Twin and Four Screw Vessels of Types 1, 2 and 3.—The great majority of self-propelled model tests show the values of K and $1\text{-}S_B$ are practically the same for different drafts. Where there has been a difference, it has as a rule been very slight.

If the hull is analyzed for K and $1\text{-}S_B$ at various drafts, the analyses would show, in most cases, a small variation in K and a somewhat greater variation in the values of $1\text{-}S_B$ (due chiefly to the varying values of $h \div H$). It, therefore, becomes necessary to know the draft at which the hull should be analyzed in order to determine its type and the value of $(B \div L) \div (1\text{-}BC)$ both of which effect the values of K, and $1\text{-}S_B$. In order to determine this draft, the following rules should be used:

Rule 1. Vessels having broad flat sterns such as destroyers, use the load water line.

Rule 2. Vessels with cruiser sterns where the load water line plane covers one-half or more of the propellers (L.W.L. "A", Sketch "B" on Chart 4A).

(*a*) When the draft is equal to or less than three-tenths of the beam ($H \div B = .30$ or less) use the load water line.

(*b*) When the draft is greater than three-tenths of the beam ($H \div B$ is greater than .30) pass a vertical line through the center of the propeller as shown on Sketch A on Chart 4A. This line intersects the hull at water line "E" which should be used as the load water line when determining the type and making the hull analysis, provided that the draft to this water line is not greater than four-tenths (0.40) or less than three-tenths (0.30) of the beam. If the draft to water line "E" is greater than four-tenths of the beam, use a load water line whose draft is equal to .40 $\times$ Beam. If the draft to water line "E" is less than three-tenths of the beam, use a load water line whose draft is equal to .30 $\times$ Beam.

Rule 3. When the load water line does not cover at least one-half of the propeller (L.W.L., "B" Sketch "B" on Chart 4A) use the deep loaded water line draft provided this draft does not exceed four-tenths of the beam (.40 $\div$ Beam) which is the maximum draft that should be used. (This applies to all types of sterns.)

Sketch A on Chart 4A shows the draft of the load water line greater than that of water line "E" (where a vertical through the center of the propeller pierces the hull). Should the draft of the L.W.L. be less than that of water line "E" and the vessel is between Types 1 and 3, it should be considered as one where the L.W.L. plane covers at least one-half of the propellers, when determining the value of K from Chart 4A, provided the vessel plots on Chart 3 as Type 1 or between Types 1 and 3 when using the draft to water line "E" for the value of H in the formula $H \div B$.

Single Screw Vessels of Types 1, 2 and 3.—As the value of $1\text{-}S_B$ varies for different drafts, the hull characteristics should be obtained at the draft for which the performance is desired when determining this value.

The value of K for all drafts should be approximately equal to that found by analyzing the hull at a draft equal to .40 × Beam when the deep loaded draft exceeds this figure. When the deep loaded draft is equal to or less than .40 × Beam, use the K value at the deep loaded draft for all conditions of loading.

Ships of Type 4.—This type consists of very shoal draft vessels fitted with tunnels in which the propellers operate.

As there is not sufficient data available to the writer, no definite rules can be laid down for determining the values of K and $1\text{-}S_B$. From what data there is available, it appears that the value of K varies from 1.80 to 2.30 and the value of $1\text{-}S_B$ is about .80 when operating in shallow water. It is not at all unlikely that the ratio of length of the tunnel to its height, the percentage immersion of the propeller and the depth of water have an influence on the values of both K and $1\text{-}S_B$.

As shown on Chart 5C the correction factors for the height of the propellers above the base line ($S.C.F._h$) for torpedo boats and destroyers are different from those of other types of vessels. As there may be occasions when it will be difficult to determine which line to use, the following rules are given as a guide, although, due to the lack of sufficient information, no definite rule can be laid down at the present.

1. Use line marked Twin Screw Types 1, 2 and 3 for the following conditions:

(*a*) When the value of L.A.B. ÷ H′ is less than 12 to 13.

(*b*) When the value of L.A.B. ÷ H′ is greater than 12 to 13 and the displacement length ratio is greater than 50 to 55.

(*c*) When the value of L.A.B. ÷ H′ is greater than 12 to 13, the displacement length ratio is less than 50 to 55 and the midship section coefficient is greater than .85.

2. Use line marked Destroyers when L.A.B. $\div$ H is greater than 13, the displacement length ratio is less than 50 to 55 and the midship section coefficient is less than .85.

HULL ANALYSIS FORM FOR OBTAINING VALUES OF "1-S_B" AND "K"

In order to expedite the work required to obtain the values of 1-S_B and K the form marked "HULL ANALYSIS" has been prepared, on the top of which is the hull data required.

If the vessel is a triple screw ship the value of H′ for the wing propellers should be the same as the mean draft H, but for the center screws the value of H′ is the height of the upper tip of the propeller above the base line. If the mean draft should be considerably greater than .40 of the beam, the hull should be analyzed at a draft equal to .40 of the beam for the wing screws,* but the actual draft (and displacement at this draft) should be used for obtaining the value of 1-S_B for the center screw.*

On quadruple screw ships where the propellers on the inner and outer shafts are to be the same, it is only necessary to obtain the average value of h.† If the power on the inner shafts is the same as on the outer and it is desired to operate all shafts at the same revolutions, it will be necessary, in some cases, to use propellers of different dimensions on the inner and outer shafts. In such cases the value of h for each should be used to obtain their respective values of 1-S_B.

TIP CLEARANCE.

In order to prevent vibration, it is necessary to have sufficient clearance between the tips of the propellers and the hull. It has been found on a large number of vessels ranging from 100 to 700 feet water line length, satisfactory results have been obtained when the tip clearance is at least one-tenth of the square root of the load water line length ($.10 \times \sqrt{\text{L.W.L.}}$) on merchant ships and on the heavier constructed naval vessels such as battleships, battle cruisers and auxiliary vessels, and thirteen-hundredth of the

* See Rules for Obtaining Values of "K" and "1—S_B".

† For obtaining the average Rpm. of outer and inner shafts. If the Rpm. of outer and inner shafts are required the value of h for each should be obtained and used for their respective shaft.

HULL ANALYSIS		
	HULL DATA	
1	L = L.W.L or L.B.P. in Feet	
2	B = Greatest Submersed Beam in Feet.	
3	H = Mean Draft in Feet	
4	Δ = Displacement in Tons (2240 Lbs)	
5	A.M.S = Area of Submerged Midship Section in Sq. Ft	
6	L.A.B = Length of After Body in Feet	
7	H' = Stream Line Height in Feet	
8	h = Height of Center of Propeller above Base Line in Feet	
9	B.C = Block Coeff. = (Δ x C) ÷ (L x B x H) = [(4) x C] ÷ [(1) x (2) x (3)] C = 35 for Salt Water = 36 for Fresh Water	
10	M.S.C = Midship Section Coeff = A.M.S ÷ (B x H) = (5) ÷ [(2) x (3)]	
11	T.C. = Minimum Tip Clearance between Hull and Tip of Propeller in Feet *	
12	α = Angle of web See CHART 4C * *	
13	A.Sta.19 = Area of Station 19 in Sq. Ft. †	
14	Sta.19 Coeff = Coeff. of Station 19 = A.Sta.19 ÷ (B x H) = (13) ÷ [(2) x (3)]	
	ANALYSIS FOR TYPE	
15	H ÷ B = (3) ÷ (2)	
16	M.S.C. Same as (10)	
17	Type from CHART 3	
	ANALYSIS FOR "K"	
18	B ÷ L = (2) ÷ (1)	
19	1 - B.C = 1 - (9)	
20	(B ÷ L) ÷ (1 - BC) = (18) ÷ (19)	
21	√L = √(1) *	
22	T.C ÷ √L = (11) ÷ (21) *	
23	Type of Stern † †	
24	K = K from CHART 4A, 4B or 4D	
25	K.C.F. = Correction Factor for K * * *	
26	K = Corrected Value of K = (24) x (25)	
	ANALYSIS FOR "1 - S_B"	
27	(B ÷ L) ÷ (1 - B.C) Same as (20)	
28	H ÷ B Same as (15)	
29	S.B.C = Slip Block Coeff from CHART 5A	
30	L.A.B ÷ H' = (6) ÷ (7)	
31	1 - S_B = 1 - Basic Slip from CHART 5B	
32	h ÷ H = (8) ÷ (3)	
33	S.C F_h = 1 - S_B Correction Factor for Height of Propeller above Base Line CHART 5C or 5E	
34	S.C.F_α = 1 - S_B Correction Factor for Angle of Web (α) CHART 5D * *	
35	1 - S_B = Corrected Value of 1 - S_B = (31) x (33) x (34)	

* For twin and quadruple screw ships. † For single screw ships of Type 2.
* * For twin and quadruple screw ships fitted with spectacle frames
* * * For twin and quadruple screw ships fitted with spectacle frames see CHART 4C
For single screw ships with Contra-rudder K.C.F. = .80 to .85
For single screw ships with ordinary stream line rudder K.C.F = .90
† † For single screw ships of Type 2 see CHART 4E

square root of the load water line length ($.13 \times \sqrt{\text{L.W.L.}}$) on the lighter vessels such as destroyers and light high speed cruisers.

When the value of $(B \div L) \div (1\text{-B.C.})$ of a vessel of Type 1 with a cruiser stern and the L.W.L. plans covers one-half or more of the propellers (water line "A" Sketch B on Chart 4A) or a vessel of Type 3 with a cruiser stern exceeding .23 a lower value of ("K" can be obtained, as shown on Chart 4A, by using a tip clearance greater than $.10 \times \sqrt{\text{L.W.L.}}$

The distance between the leading edges of the propeller blades and the after edge of the struts (or web on ships with spectacle frames) should be as great as possible, as this has an influence in preventing vibrations.

Two more sections of this paper will be published in the next two issues of the JOURNAL.

NOTES.

INDEX TO NOTES.

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MILLION-VOLT INDUSTRIAL X-RAYS.
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Sparks from Aluminum Painted Steel.
—Metals and Alloys, January, 1942.

BEARINGS AND LUBRICATION.

This paper was presented at a meeting of the Petroleum Division of the Metropolitan Section, The American Society of Mechanical Engineers, held in New York City on January 22, 1942. It has since been published in Mechanical Engineering for April, 1942. The author, R. J. S. Pigott, is associated with the Gulf Research and Development Company, of Pittsburgh, Pennsylvania. He maintains that many bearing troubles traceable to design can be fairly easily avoided by making use of theories established as long ago as 1886.

In these days, the lubrication of certain classes of bearings, particularly those employed in heavy-duty, high-speed, internal-combustion engines and in automotive and aeronautic equipment generally, has become in many cases acutely troublesome. Apparently a good deal of this difficulty is due to the fact that the theory of the bearing has not been fully developed to practical application although the major basis of this work was Osborne Reynolds' theory, published in 1886; and even such theory as we have is receiving scant attention, especially among automotive designers. Observation has shown that a great deal of this trouble can be avoided by making some real use of what we already know about the operation of bearings. The theoretical treatments are now almost exclusively in the hands of mathematical physicists and are too much for the average engineer who has not mathematics enough to understand what these specialists write.

Principles of Ordinary Bearing.

A brief survey of the basic principles on which an ordinary bearing works may be illuminating. Figure 1 shows the ordinary behavior of a bearing which has no lubricant in it. When at rest, it lies at the bottom of the sleeve with the center line of the shaft vertically under the center line of the shell. When rotation starts clockwise, the bearing tends, by way of friction at point a, to climb up the side of the shell. Such a characteristic can be seen on a lawn-mower front roller bearing, particularly as this bearing is not usually lubricated. The moment a lubricant is added to the bearing, however, the situation changes in a surprising way. Figure 2 shows an unloaded lubricated, journal during rotation. This would be the case, say, of a grinder or a spindle transmitting only rotary motion, running at high speed. The journal tends to come very close to being centered in the shell with a film of liquid very nearly uniformly distributed all around it. This bearing, like any other, will develop heat, owing to the rapid shearing of the oil films over each other. Figure 3 shows this condition. The velocity of the oil at the journal is the same as the surface of the journal since there is no slip between the oil and shaft. But the slip develops as we pass through the clearance space to the shell until by the time we reach the shell the oil is stationary and has no relative motion to the bearing shell. The work done and consequently the heat developed can be quite accurately computed and the formula takes the general form

$$W = C \mu a v^2 / b$$

where W = work, ft-lb

μ = viscosity, ft-lb-sec units—Reyns

a = area of rubbing surfaces, sq ft

v = rubbing velocity of bearing, ft per sec

b = clearance, ft

C = a constant

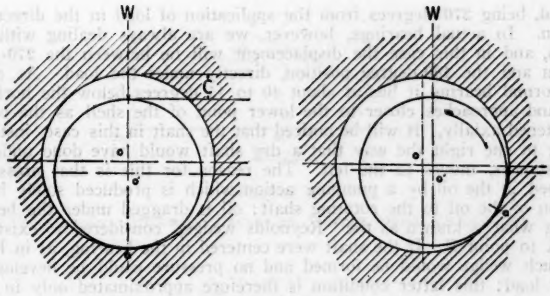


FIGURE 1.—(LEFT) DRY JOURNAL AT REST IN BEARING; (RIGHT) DRY JOURNAL DURING ROTATION.

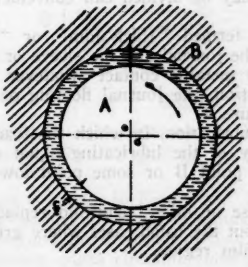


FIGURE 2.—UNLOADED LUBRICATED JOURNAL DURING ROTATION.

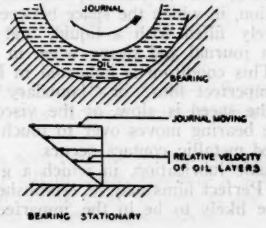


FIGURE 3.—SHEAR FORCES IN UNLOADED BEARING.

As soon as material load is applied to the bearing, the condition becomes that shown in Figure 4. This figure shows the conditions for a bearing of infinite length (which of course we do not have), and theoretically the center line of the shaft should move exactly at right angles to the line of

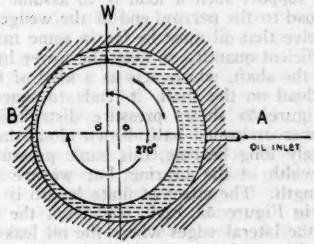


FIGURE 4.—LUBRICATED JOURNAL IN RUNNING POSITION.

the load, being 270 degrees from the application of load in the direction of rotation. In actual bearings, however, we are always dealing with finite lengths, and in this case the displacement will lie between the 270-degree position and the 180-degree position, directly under the load. As a rule, in a normal bearing it lies at about 40 to 45 degrees below the horizontal plane and approaches closer to the lower point of the shell as the bearing is shortened axially. It will be noticed that the shaft in this case, instead of moving to the right the way that a dry shaft would have done under the circumstances, moves to the left. The reason for this is that pressure is developed in the oil by a pumping action which is produced solely by the adhesion of the oil to the rotating shaft; oil is dragged under the bearing, forming what is known as the "Reynolds wedge," considered to exist from point A to point B. If the shaft were centered in the bearing, as in Figure 2, no such wedge would be formed and no pressure could be developed to support load; this latter condition is therefore approximated only in high-speed bearings running very lightly loaded.

Types of Lubrication.

The divisions into which lubrication may be divided can conveniently be stated as follows:

1 Perfect film lubrication, sometimes termed "thick film" or "flood" lubrication, in which the space between the journal and the sleeve or box is completely filled with a liquid, and no metallic contact whatever occurs between journal and sleeve when in motion, the journal floating on liquid film. This condition is indicated in Figure 4.

2 "Imperfect film" or "boundary" lubrication, in which the load is so great, the speed is slow, or the viscosity of the lubricating liquid so low that the bearing moves over to touch at point B or some point lower and a limited metallic contact occurs.

3 Plastic lubrication, in which a grease or other semisolid replaces the liquid. Perfect films may be established but the usual cases where grease is used are likely to be in the imperfect film region.

The Reynolds Wedge.

The reason a liquid film can "float" a loaded journal was first clearly shown by Prof. Osborne Reynolds in 1886 and his analysis was based on Beauchamp Tower's experimental data of 1883 and 1884. Petroff in 1883 had recognized the connection between viscosity and friction, but Reynolds gave us the underlying theory of liquid lubrication.

In order for a shaft to be floated on liquid, it is obvious that pressure must be developed in the film of liquid underneath the journal. Reynolds demonstrated mathematically that the only way in which a film can develop lateral pressure to support such a load is to assume a wedge shape, with motion from the broad to the narrow end of the wedge. Looking at Figure 4, it is easy to perceive that oil must be fed in some manner, as for instance at groove A, in sufficient quantity to keep the lower half full; then as it is dragged around by the shaft, which acts as a kind of friction pump, and is compressed by the load on the shaft, it tends to squeeze out at the sides of the bearing. Figure 5a shows pressure distribution around the shaft developed in the lower or loaded half, drawn along the center of the bearing. For an infinitely long bearing, this same pressure would exist completely across the width of the bearing, but we are concerned only with bearings of finite length. The effect of finite length is to drop off the pressure hump, shown in Figure 5a, each way from the center line until the pressure is zero at the lateral edges where the oil leaks out. The pressure distribution across the width of the bearing at the region of maximum

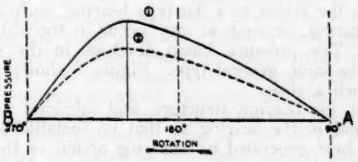


FIGURE 5a.

FIGURE 5a.—POSITIVE PRESSURE DISTRIBUTION OF FIGURE 4.

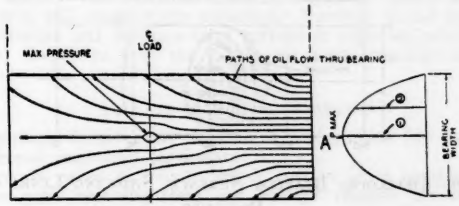


FIGURE 5b. FIGURE 5c.

FIGURE 5b.—DEVELOPMENT OF LOWER HALF OF BEARING OF FIGURE 4.
FIGURE 5c.—FILM PRESSURE DISTRIBUTION ACROSS BEARING (FIGURE 5a).

pressure is shown in Figure 5c. It is seen that the pressure distribution in such a bearing is a kind of mound or hill in shape, reaching a peak somewhat beyond the direction of load and falling away to zero at the entering and leaving edges circumferentially and at both edges axially.

Theoretically, there would be a negative or suction pressure in the top half of the bearing equal in amount to the positive pressure in the lower half, but here the mathematical analysis fails to meet the facts for several reasons. The most important is that a negative pressure of only one atmosphere could be attained. Some slight vacuum does often occur; it is known that in "starved" bearings some air may be drawn in on the unloaded side. Such negative pressures as actually develop are too slight to affect the load capacity materially and in practical bearing design they are completely disregarded.

The approximate flow path of the lubricant when fed by groove at A is shown in Figure 5b.

The important thing in designing a bearing is to determine the amount of eccentricity which will occur under given load per square inch of projected bearing area, rubbing speed, viscosity, and clearance. It is obvious that for a given set of these conditions, the load can be increased to a point where the eccentricity becomes so great that the shaft touches the shell. This condition starts to break down the film of liquid lubricant, metallic contact begins to occur, and the state becomes that of "imperfect film" or "boundary" lubrication. If the load is further increased so that general metallic contact occurs, complete rupture of the film takes place. Except for a few special cases where the materials are selected for the purpose and the rubbing speeds are very low, abrasion and seizure will generally result. Figure 6 shows the actual pressure measurements of the pressure hump in a bearing, developed to show the plot.

Figure 7 shows the action in a flat-type bearing, such as a Kingsbury or Michell thrust bearing, or such as may occur in the piston of an ordinary gasoline engine. The pressure hump develops in the same manner and leakages are of the same general type. Figure 8 shows measured pressure distribution on such a shoe.

It is plain that the bearing structure and lubricant together must (1) support the journal in the bearing so that no metallic contact occurs and (2) remove any heat generated by shearing action of the lubricant or the displacement of the shaft from the center position, which also adds to the work of the bearing.

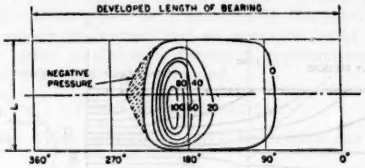


FIGURE 6.—DEVELOPED BEARING SURFACE SHOWING LINES OF EQUAL PRESSURE.

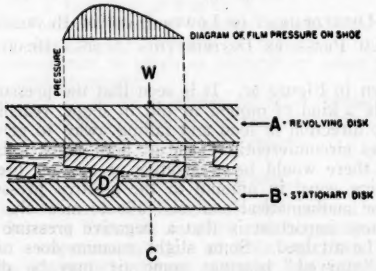


FIGURE 7.—PRINCIPLE OF KINGSBURY OR MICHELL THRUST BEARING.

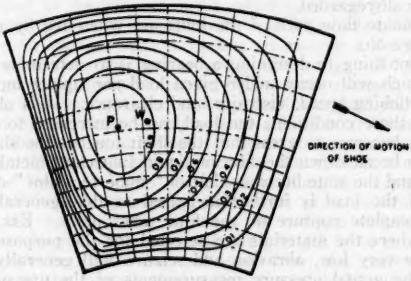


FIGURE 8.—PRESSURE DISTRIBUTION OVER THRUST SHOE.

A little inspection will show that those portions of the bearing from horizontal to 45 degrees below horizontal are of very little value in any case in supporting the load, since they have a very small vertical component of pressure and, for this reason, partial bearings such as half boxes or 120-degree boxes are often used and will carry about the same load under the same conditions of speed, viscosity, and clearance as a full bearing. In fact, at high speed, the bearings will carry more, because the area of close clearances which generate heat in the oil by rapid shear is decreased and the heat generated is less, since the oil shear is reduced.

The mathematical theory as developed by Reynolds has never been reduced to a rigorous solution for a finite bearing until very recently. For many years the work of Kingsbury, Howarth, and Needs in graphical methods of solution had rendered Osborne Reynolds' theory available for design, but it is still insufficiently employed. Recently at the Gulf Laboratory, Drs. Muskat and Morgan have arrived at rigorous solutions of the mathematical theory to give the values for finite bearings such as are actually used.

The Case of Actual Bearings.

Actual bearings depart from the simple assumptions for the Reynolds theory in several important particulars:

1 Bearings are never of infinite length, and in finite bearings the complicated flow path resulting from end leakage renders the mathematical solution correspondingly difficult.

2 The heat generation from oil shear and shaft displacement lowers the viscosity of the lubricant as it passes through the bearing and, again, not in a simple manner.

3 For many cases, the shaft and bearing housing together with the lubricant are the only means of heat removal, and this may be irregular with regard to the bearing surface. In most internal-combustion-engine crankshafts and many other cases, the bearing parts, such as connecting rods and crankcase webs, actually bring additional heat to the bearings from the combustion chamber, so that the only remaining source of cooling is the lubricant itself.

4 Many types of grooving and oil feed are employed. Every change of grooving from the simple assumptions made and shown in the foregoing figures causes a change in flow path and pressure distribution, and consequently a change in load capacity. We have almost no good test work on the effect of grooving on the load-carrying capacities, or the effect of increasing feed pressure. The general conclusion from such experiments carried out so far is that the load-carrying capacity can be raised by increasing feed pressure, but the amount is considerably dependent on location of feed, and the load-carrying capacity can also be raised or lowered by change of grooving.

5 Faulty alignment of shaft or bearing, or distortions under load, have a very pronounced effect on safe load and heat generated and cannot be analyzed directly. In the test machine, we may be able to develop, without any distress in the bearings, loads up to 3000 or 4000 psi, but in very few operations in which misalignments are taking place can any such figures be ordinarily attained.

The Function and Effect of Oil.

A bearing such as those described could be lubricated with a mixture of sugar and water, milk, or in fact any lubricant that has sufficient viscosity, but the reason we use oil is that it is almost the only commercial material obtainable in large quantities and at relatively low cost that will stay in the

same condition unchanged chemically over long periods of time, even when considerably heated. Figure 9 shows the effect of the character of the oil. Line a shows the bearing operated with sufficient cooling from some source so that all heat is removed as fast as it is generated and there is no change in the temperature of the oil passing through the bearing. Line b shows effect when an oil of high viscosity index is allowed to run through, with no other cooling. Reduction of viscosity going through the bearings reduces the pressure developed. Line c shows the effect when an oil of the same initial viscosity at room temperature, but lower viscosity index, is employed. This is an oil in which viscosity would reduce more rapidly with temperature than the oil used for line b. It is therefore plain that, if the oil is to be depended upon for removing heat, then the highest load-carrying capacities can be obtained in many cases by the use of oil of high viscosity index. However, if the bearing is cooled by other means, such as water jackets, then it is possible to take the cooling function away from

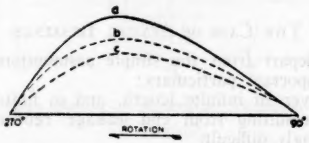


FIGURE 9.—FILM-PRESSURE DIAGRAM.

the oil, and an oil of low viscosity index can be quite successfully used. There are other places in an internal-combustion engine where the differences in properties of these two kinds of oils have a bearing on which one is employed. It may be borne in mind that the viscosity of an oil of high viscosity index varies inversely about as the 2.5 power of F. temperature; a low-viscosity-index oil, as the 3.0 power or more; water varies as about the first power.

EFFECT OF GROOVING.

The effect of grooving has been discussed at length over a long period of years, but we have acquired very little test proof of any arguments. One such case that has been investigated is the annular groove commonly employed for connecting-rod bearings in many automobile engines. This is shown in Figure 10. It has often been stated by those thinking only in terms of the Reynolds theory that such a bearing should carry less load than before the annular groove is applied, the reason being that since the area is reduced and two additional leakage edges are afforded—that is, the bearing is converted into two of a little less than half the length each—the pressure hump should be reduced. Dotted line a at the right of the figure shows the ungrooved bearing pressure; dotted line b, that of the load-carrying capacity, supposing no change in viscosity occurs because of heating. But what actually happens is that when the annular groove is employed, the capacity for flow is more than doubled and may in many cases be tripled. This means that the reduction of viscosity for the same rate of viscous shear is much less because the temperature of the oil rises only one half or one third as much, and the net effect is generally an increase of load capacity which may go as high as 80 per cent for some cases. This has been proved by test.

The general opinion on grooving, as yet insufficiently supported by definite test, is that the oil supply should preferably be on the unloaded side near the thick end of the Reynolds wedge (A-B, Figure 4), and should not be across the loaded area. While Figure 10 appears to transgress these rules, as already explained, actually it does not. It merely converts the bearing into two shorter ones with full end feed. The advantage of this bearing is its ability to flow more oil than any other grooving and therefore to cool well.

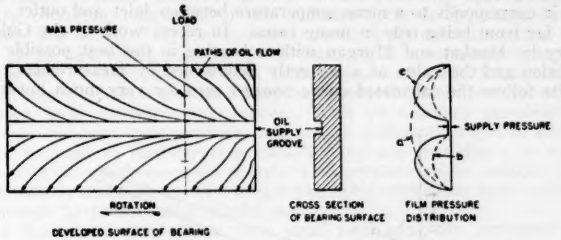


FIGURE 10.—CONDITIONS AFFECTED BY OIL GROOVE.

In recent years, some bearings have been produced that have operated very successfully at extremely high bearing pressures—as high as 5000 psi. These are mostly confined to large bearings for rolling-mill necks; in this case the end leakage has been partially suppressed by means of packing devices at the edge of the bearings which in effect converts the bearing into something approximately like a bearing of infinite length. Moreover, in these bearings oil is flowed through so copiously that rise in temperature is only a few degrees. Also, the bearing is so set that the effects of misalignment and distortion are substantially wiped out.

To illustrate how important these effects can be, consider a rotary pump of a simple three-piece design, in which the bearing loads are quite high, which consists of a center housing and two end plates into which rigid bushings are pressed. It would take a week to bring this pump up to 600 psi using babbitt bearings, adding the load in small increments. If the pump were then dismantled and put together again, there would be sufficient change of alignment so that the conforming and wearing-in process would have to be completely repeated—that is, the only reason the bearing would carry the load at all is that the babbitt used would mush away or "conform" until it would assume a position in line with the shaft for whatever loads were occurring. Recently, application of bearings supported flexibly to follow the shaft freely resulted in the ability to bring the pump up to full pressure in 15 min with new bearings. Incidentally, this was done on water, and loads in excess of 470 psi were successfully carried with as poor a lubricant as water.

THEORIES OF FILM BREAKDOWN.

Sommerfeld, in reducing the Reynolds expressions by definite integration, foreshadowed an operator—the product of viscosity and speed in revolutions divided by unit load per square inch—that is receiving considerable attention. Some years ago Hersey proposed the plot of friction coefficient against this operator and this plot has some quite useful properties, as shown

in Figure 11. This method of plotting, however, has also some limitations. The plots for different bearings do not coincide; the friction coefficient is a variable with change of oil, of bearing type and grooving, and with bearing surface condition and cooling means. Nevertheless, it has been considerably used by many engineers as a means of showing the point at which "boundary" lubrication is reached, that is, when the film tends to become unstable and the bearing approaches seizure.

The weak factor in this plot is that we are ordinarily unable to determine accurately the value of μ, the mean viscosity, and we have usually assumed that it corresponds to a mean temperature between inlet and outlet. This is very far from being true in many cases. In recent work at the Gulf Laboratory by Muskat and Morgan with a bearing in the best possible surface condition and the value of μ correctly determined by measurement, the test results follow the calculated curve toward zero for very much lower values

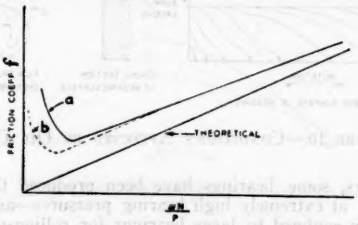


FIGURE 11.—RELATION BETWEEN COEFFICIENT OF FRICTION AND $\mu N/P$.

(μ = mean absolute viscosity, centipoises; N = revolutions per minute; P = load on projected area, psi; f = coefficient of friction.)

of $\mu N/P$ than is usually taken as the critical point. A bearing in relatively rough surface condition may show some such curve as a, Figure 11; when run in, lapped smooth, or otherwise reduced to very low roughness, with other conditions the same, the critical point will move down more or less like curve b. Or, by adding certain materials to the oil, generally known as "oiliness agents," the same shift can be obtained without altering the bearing surface.

In the perfect film region of lubrication, the only properties of the lubricant involved are viscosity and, of course, change of viscosity with temperature, but it has been known for a long time that certain materials could carry a bearing farther into the boundary-lubrication region without failure; that is, change would be, in Figure 11, from a to b. This quality, christened "oiliness," is generally defined as that property in an oil which reduces the coefficient of friction at or near the boundary-lubrication condition. This is about all we can say about it because nobody knows how it works. Two theories are advanced for the action of oiliness agents. The first theory is that there is a selective adsorption on the surface of the metal, which consists chiefly of those constituents in the original oil having oiliness properties. When the clearance, because of increased load, becomes sufficiently small so that these two adsorbed films meet, it is the same as if a more viscous lubricant, like a grease, were used, and the bearing will carry additional load. This theory appears to be contrary to some of the facts. If a lubricant of higher viscosity should thus appear, the friction

coefficient should increase, instead of decrease, for an appreciable distance. The physicists hold that in any case this film can be only one molecule thick, and we can't make bearings that smooth.

The other theory is that another property of the oils used may account for the action. It is known that the viscosity of petroleum oils increases with pressure, and at high pressures the increase may be quite rapid. In general, however, the oiliness agents themselves, such as lard oil and stearic acid, tend toward increased viscosity rather rapidly with pressure and become solids at more moderate pressures. In a bearing approaching the boundary condition, roughness of the surfaces becomes important because the "hills and valleys" of the surface (actually only a fraction of a thousandth of an inch high) may be of the same value as minimum clearance, in which case the high spots on each surface tend to hit and break the film. At these points, which represent a very small percentage of the total surface, very high local pressures may occur. With an ordinary petroleum oil, increase of viscosity with pressure may not be sufficiently rapid to afford a material increase in load-carrying capacity, but any ingredients in the oil the viscosity of which increases sharply with pressure might convert it into a more viscous material or even grease-like solid which has flow resistance high enough to prevent film breakdown.

Since these local points have very small total area, the increased shear resulting from local increase in viscosity is too small to be detected and the friction factor continues to decrease with reduction of $\mu N/P$ to some lower value where, because of further reduction of minimum clearance, there are many more high spots ready to touch, local pressures become still more severe, and the little local grease spots can no longer furnish enough increase in viscosity to support the load. The bearing is then in the same condition as it was with the plain mineral oil with no oiliness agents present, and is ready for failure.

Both the foregoing theories lack proof. Kingsbury proved 20 years ago that down to a clearance of 0.000025 inch the bearing still follows the hydrodynamic theory. Later experiments have proved that this clearance can be as low as 0.00001 inch. We have enough test data to show that some of the modern methods of bringing a finely finished bearing down to 3 to 7 microns does increase the load-carrying capacity quite noticeably.

Most of us believe from experience that the smoother the bearing, the higher the load capacity. Following another school of thought, one of the larger automobile companies is deliberately roughening the bearings to increase the load-carrying capacity and it is extremely curious that two such opposite ideas should persist in these times. What is probably happening in these roughened bearings is that conformability of the bearing to the normal distortions is increased. It is probable that the theory has been advanced from the known fact that piston rings with relatively rough-turned surfaces will seat more quickly than dead-smooth rings. One must remark, however, on the fact that people who are advocating roughening have presented absolutely no test data to prove that the bearing will actually carry more load.

Use of Babbitt.

In perfect film lubrication, the materials in the shaft and bearing shell appear to be of no importance whatever, but become important during periods of starting or stopping, or under heavy loads at low speed when borderline condition may occur. The widespread and successful use of babbitt metal is explained chiefly by the fact that it may partially fail without completely failing as a bearing, and in general it may fail without damaging the shaft. On the other hand the melting point of babbitt is

relatively low and its strength decreases rapidly with temperature. Consequently, in internal-combustion engines, particularly where there has been insufficient oil flow over the bearing, temperatures have been brought up to such a point in the last few years that fatigue failures have been frequent. This situation has led to the use of other alloys such as cadmium silver, cadmium nickel, and copper lead. These materials have much higher strength to withstand the bearing temperatures now subsisting but have introduced many other troubles. Since these alloys are much harder and stronger than babbitt, conformability is much less and the danger of ruining the bearing during break-in is much higher as a rule. Furthermore, in general these alloys require larger clearances than babbitt and this may improve the situation by allowing a more copious oil flow over the bearings. The corrosion troubles with these alloys, particularly copper lead, have been so bad in the last two or three years that there is now a reversion to babbitt in another form. Steel backs covered with a coating of very moderate thickness of some intermediate material like copper nickel or powdered metal, and then with a thin skin of babbitt 0.005 to 0.008 inch thick for the bearing surface, allow the babbitt to carry as high loads as any of the copper-lead bearings, while retaining desirable conformability and other advantages characteristic of babbitt. Babbitt also eliminates the corrosion.

The author has felt for some time, and is now gathering considerable evidence in support of his belief, that the correct way to have cured troubles with babbitt bearings in internal-combustion engines was to improve the lubricating system and not necessarily change the alloy. He has found that the automotive engineers have paid very skillful attention to design of connecting rods, pistons, crankcases, and shafts for strength but have done almost no real designing of the lubricating system, which is just as susceptible to calculation within quite close accuracy as any other part of the engine. They are now beginning to realize that besides piston and steel shaft they have also a third material, oil, which must be treated as having breaking strength and as being capable of overloading just as metal is.

An illustration from one case used in proving the foregoing argument a year ago might be pertinent. Figure 12 shows the connecting-rod bearing shell of a popular light car of some years ago that had very serious trouble with the main bearings in a small percentage of its output. These were usually confined to cases of long-distance hard driving, and a complete set of main bearings, including connecting rods, might be ruined after anywhere from 300 to 3000 miles of driving. The connecting-rod big end, Figure 12, is lubricated in the usual way by means of supply through a hole in the shaft and an annular groove; but in addition it has a triangular groove, shown enlarged in the insert at the right, which has an astonishingly high oil-flow capacity. As a matter of fact, this groove actually flowed off from 90 to 93 per cent of the total oil fed to the bearing by the annulus, so that with a total supply to each connecting-rod bearing of around 0.65 to 0.75 gallons per minute, the actual flow over the bearing surface, for both lubrication and cooling, was from 0.021 to 0.040 gallons per minute, depending on speed and kind of oil employed. It was recommended to the designers that the triangular cross groove which was intended for removing dirt should either be cut out or reduced to a surface depression only a few thousandths of an inch in depth. The actual solution adopted by the designers was to cut out the annular groove. In this case it was found that the supply pressure of oil reaching the connecting rod was automatically jumped about 15 to 20 per cent and the flow over the bearing was increased about 60 per cent, largely as a result of eliminating the triangular dirt groove. These bearings then were out of trouble.

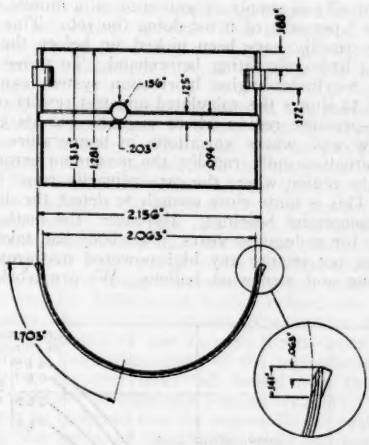


FIGURE 12.—CONNECTING-ROD BEARING OF A 1935 LIGHT EIGHT ENGINE.

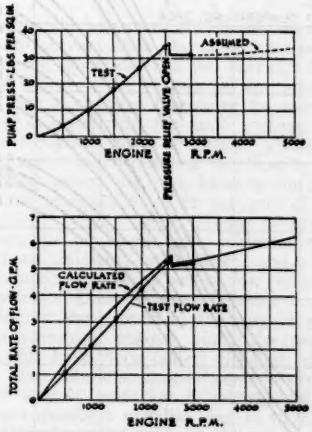


FIGURE 13.—COMPARISON BETWEEN TEST AND CALCULATED DATA.

(*Top*: Pump pressure vs. engine rpm. *Bottom*: Total rate of flow through engine vs. engine rpm.)

(1935 light eight engine; S.A.E. 20 oil, viscosity index 105; ambient temperature 280 F.)

But it does seem silly to supply ½ gallon of oil a minute to a bearing and then use only 6 or 7 per cent of it for doing the job. This kind of error in design could very readily have been picked up before the car got on the market by doing a little calculating beforehand. To prove that so complex a problem as an 8-cylinder-engine lubrication system can be successfully computed, Figure 13 shows the calculated and test results on the actual engine. In the low-pressure region where viscosity has its greatest effect on the computed flow and where variations in temperature of the bearings would alter the situation quite rapidly, the maximum error is not over 10 per cent, and in the region where the car ordinarily runs, the error is only 3 or 4 per cent. This is quite close enough to detect the difference between successful and unsuccessful bearings. Moreover, the method of calculation has been available for at least 40 years, if anybody had taken the trouble to use it; and it does not require any high-powered mathematics, although it is undoubtedly long and somewhat tedious. We are working now to see

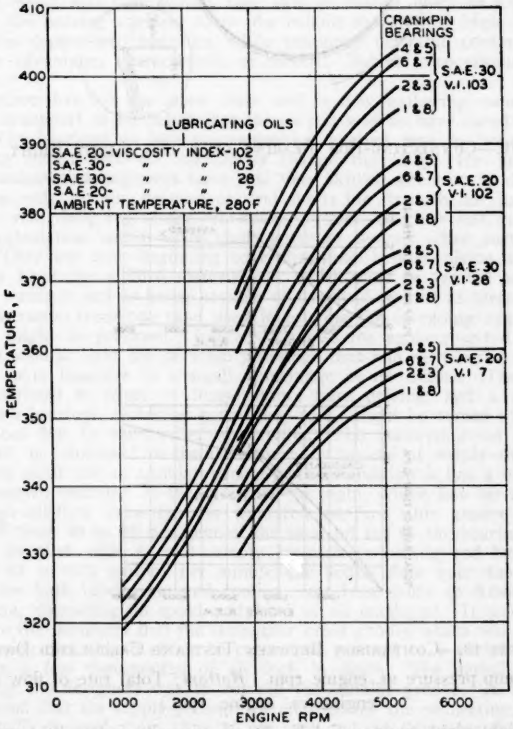


FIGURE 14.—CALCULATED MAXIMUM CRANKPIN-BEARING TEMPERATURES PLOTTED AGAINST ENGINE RPM.

if we can devise some short cuts for analyzing engines in this manner to spot weak places in the lubricating system. This calculation has already been carried out on two popular cars and one high-speed Diesel engine and, it appears, will afford a valuable means of checking the lubricating design of any type of engine.

To illustrate further how bad the condition in this particular engine with the original grooving was, Figure 14 shows that the temperature of some of the bearings in this car were running as high as 400 F., and this situation was badly aggravated by the fact that the crankcase was running altogether too hot. The crankcase, which is still used in most gasoline engines for the entire cooling of the oil, is a very poor cooler. It is interesting to note that at the present time the car which was using cadmium-silver bearings when in trouble, has since returned to babbitt and has had no trouble from bearings for four or five years. In this case the whisper got around in the trade that the cheap oils, mostly of low viscosity index, would work in this car, and the high-priced, highly refined oils would not. The reason is simply that what was needed was greater flow over the bearings, and the viscosities of the oils of low viscosity index were reduced by temperature effects much more than those of the high-viscosity-index oils so that about 30 or 40 per cent more oil flowed over the bearings. This greater flow was sufficient to eliminate bearing trouble.

Of course it must be observed that the engine did not require oil of higher viscosity to carry the loads, or the trouble could not have been cured by this means.

Cooling of Oil Neglected.

Two designs of engine which are still on the market retain the old jet oil system. These, in hard service, show the same kind of troubles as in the example mentioned in the foregoing, and the bearings do not last long because of the very low flow over the bearing surface proper, and because the cooling which in this case is supplied largely to the exterior of the big end of the connecting rod is insufficient for good cooling.

In high-speed Diesel engines there is another problem which is mainly centered around the piston. Much more of the blow-by in a Diesel engine is air than it is exhaust gases. In fact, blow-by during the compression stroke is wholly air and probably 60 to 70 per cent of the blow-by on the explosion stroke is also air. Since Diesel pistons generally run hotter than gasoline-engine pistons, the oil is not only subjected to this higher temperature but also to an oxidizing atmosphere, and ring sticking is the major trouble. Naphthenic oils, which are of low viscosity index, have in general proved desirable for Diesel engines because on the hot piston they burn off clean and do not leave much in the way of gummy deposits and hard carbon; but in two-cycle engines, since the temperatures are still higher, the paraffinic oils have done better, although they still give some trouble. This is one place where so-called "compounded" oils have proved useful. An inhibitor against oxidation is added, and also in general a detergent—which means usually a metallic soap to wash the oxidized products, as they accumulate, down into the crankcase.

It is quite possible, as internal-combustion engines are designed for higher mean effective pressure and consequently higher horsepowers per cubic inch, that it will be necessary to adopt oil cooling of the piston generally to keep the temperatures low enough to avoid this kind of trouble. In only one high-speed Diesel at the present time is the piston cooled by oil, and this does not appear to be a good job because the oil is not being applied to the place that needs cooling the most.

Examples of Copper-Lead Bearing Surfaces.

The following figures indicate some of the conditions that are plainly evident in copper-lead bearings. Figure 15 shows the uniform smooth condition of a new bearing as supplied by the manufacturer. This surface is smooth to 4 to 7 microinches roughness; so good that there is no change of appearance with 90 degree change in lighting. Figure 16 is a section from a connecting-rod bearing, rod side, used in a light car. Lead is wiped over the surface. With lighting at 90 degrees, the smooth bearing areas are dark in one case and light in the other. Some corrosion and scoring from dirt are evident. The bearing is ready for replacement but it has not failed. Figure 17 shows the cap side where there is severe scoring. Figure 18 shows a line-bored shell with a roughness of 200 microinches.

Figure 19 is a 500 magnification of a new bearing; the gray lead shows uniformly. Figure 20 shows a bearing in which the lead has been sweated out by high speed and close clearance. Figure 21 shows a broached bearing, indicating plainly the distortion of copper matrix and lead at the surface; it is quite smooth otherwise. Figure 22 shows a bored bearing such as is advocated by one motor company at present. The only feature that can be comprehended is that the ridges crush easily and allow a little of the conformability of babbitt in a hard material. Figure 23 shows the ordinary view of new reamed and lead-sweated bearings from an actual test to produce lead sweating. It represents close fit, high speed, fairly high load. Figure 24 shows a new bearing and the coarse grain found in some cases. There is quite a wide difference in makes of copper-lead bearings. Figure 25 shows how lead has been removed by corrosion. Note how similar this condition is to that of sweating. Figure 26 shows cold corrosion. Both lead and copper have been removed and black copper sulphide is present, a product not found in hot-corroded bearings. Figure 27 shows the beginning of fatigue failure in an uncorroded copper-lead bearing. Figure 28 shows another bearing in the same engine, advanced to full fatigue failure. It is evident that fatigue failures occur also in copper lead.

A corroded bearing, a lead-sweated bearing, and a fatigued bearing may look much alike when removed from the engine. It is only in the earlier stages, before failure occurs, that the type of failure is clearly evident.

Troubles Avoided by Attention to Design.

There are some methods by which conditions like this can be cured before an engine goes into production, and it is odd that some of them have not been used long ago. It has been possible, as pointed out, to calculate the flow through the oiling system to every bearing. It is also possible to calculate the heat developed in the bearings so that the maximum temperatures can be predicted with good accuracy. There is nothing abstruse about it; the calculation is rather long and tedious, but it pays very well in saving faulty designs from getting on the market. We have calculated several commercial engines and find weak spots in all of them which can be easily remedied when it is known just what the flows are. Much to the surprise of many designers, it is found that the oil flow through an engine is in general taking place in the viscous region of flow rather than in the turbulent region, that is, the flow is directly proportional to pressure, or nearly so, instead of being proportional to the square root of pressure. In general, with the bearings in ordinary condition, the highest resistance in the system is usually the bearings themselves, which also are always in viscous flow. The author hopes soon to publish a paper on the methods of calculation so that they can be readily used by everybody.

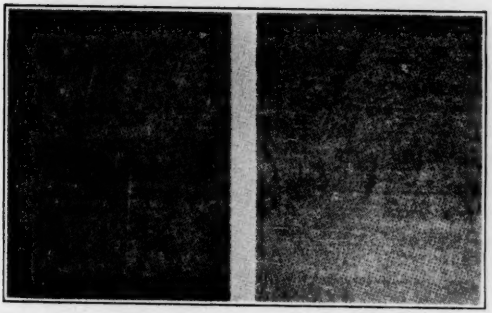


Figure 15 is a section of a new copper-lead connecting-rod shell. (Note the smooth texture in comparison with that of the other figures. Also note that change in direction of lighting does not change appearance of surface.)

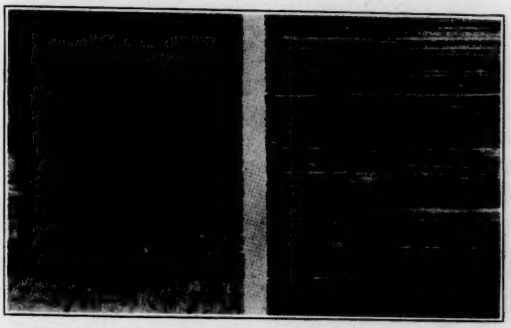


Figure 16 is a section of a copper-lead bearing shell which formed the rod half of the bearing. Lead smeared over the surface of this bearing is indicated in the left figure by the light portions. The somewhat darker portions of the left figure represent smooth bearing surface. With lighting direction changed by 90 deg, the smooth bearing surface is lighter in color in the figure at the right.

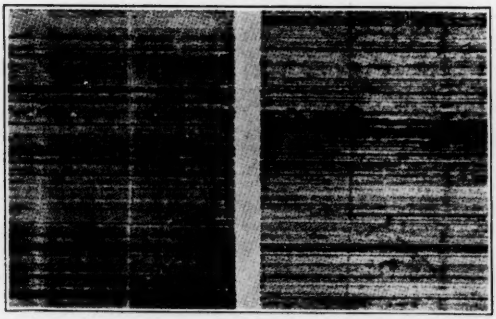


Figure 17 is a section of a copper-lead bearing shell which formed the cap half of a connecting-rod bearing. Note severe scoring in figure on the left. With lighting direction changed by 90 deg in the figure on the right the scoring is brought out less sharply.

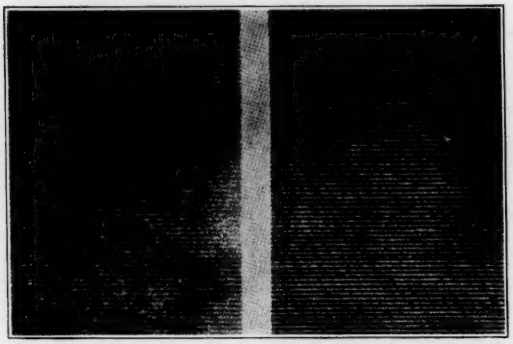


Figure 18 is a section from a copper-lead connecting-rod-bearing shell which had been line-bored. In the figure at the right, note the ridges caused by the line-boring. The figure on the right shows the effect of changing the lighting direction by 90 deg. The ridges caused by line-boring are clearly seen and the tears along the ridges are brought out more sharply.

FIGURES 15, 16, 17, 18 SECTIONS OF COPPER-LEAD BEARING SHELLS.
(In each case the lighting of the example at the right is at 90 deg to that of the one on the left.)

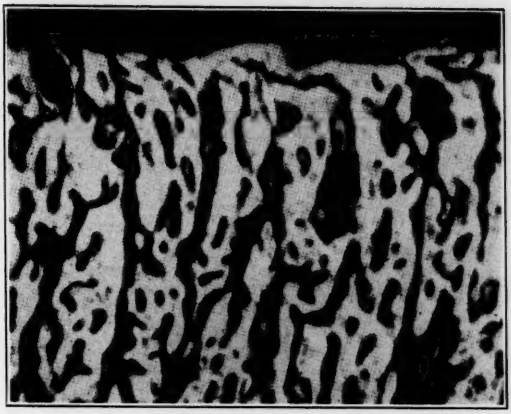


FIGURE 19.—NEW COPPER-LEAD BEARING, IN WHICH THE GRAY LEAD SHOWS UNIFORMLY, ×500.

FIGURE 20.—COPPER-LEAD BEARING SHOWING LOSS OF LEAD SWEATED OUT OF SURFACE AND COMPACTING OF COPPER MATRIX, ×500.

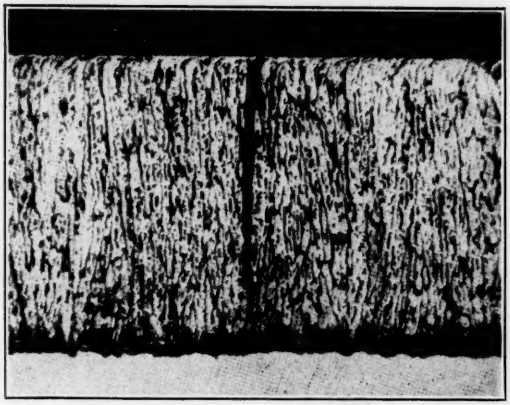


FIGURE 21.—BROACHED BEARING SHOWS DIRECTION OF BROACHING.

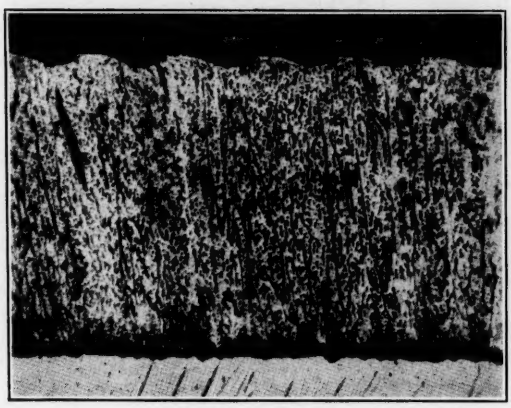


FIGURE 22.—COPPER-LEAD BEARING BORED WITH A FLY CUTTER.

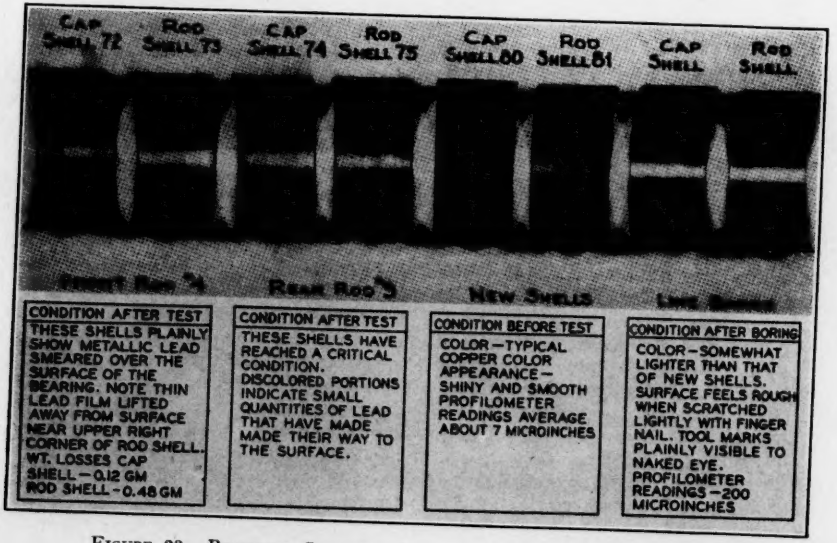

CAP SHELL 72
ROD SHELL 73
CAP SHELL 74
ROD SHELL 75
CAP SHELL 80
ROD SHELL 81
CAP SHELL
ROD SHELL
NEW SHELLS
CONDITION AFTER TEST
THESE SHELLS PLAINLY SHOW METALLIC LEAD SMEARED OVER THE SURFACE OF THE BEARING. NOTE THIN LEAD FILM LIFTED AWAY FROM SURFACE NEAR UPPER RIGHT CORNER OF ROD SHELL. WT. LOSSES CAP SHELL—0.12 GM ROD SHELL—0.48 GM
CONDITION AFTER TEST
THESE SHELLS HAVE REACHED A CRITICAL CONDITION. DISCOLORED PORTIONS INDICATE SMALL QUANTITIES OF LEAD THAT HAVE MADE MADE THEIR WAY TO THE SURFACE.
CONDITION BEFORE TEST
COLOR—TYPICAL COPPER COLOR APPEARANCE—SHINY AND SMOOTH PROFILOMETER READINGS AVERAGE ABOUT 7 MICROINCHES
CONDITION AFTER BORING
COLOR—SOMEWHAT LIGHTER THAN THAT OF NEW SHELLS. SURFACE FEELS ROUGH WHEN SCRATCHED LIGHTLY WITH FINGER NAIL. TOOL MARKS PLAINLY VISIBLE TO NAKED EYE. PROFILOMETER READINGS—200 MICROINCHES

FIGURE 23.—BEARINGS SHOWING LEAD SWEATING FROM COPPER-LEAD ALLOY AND ROUGHNESS FROM REAMING.

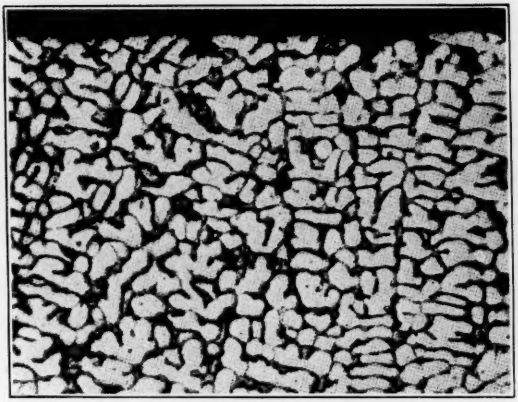


FIGURE 24.—AN UNCORRODED COPPER-LEAD BEARING.
(The lead, the gray structure, extends to the surface of the bearing.)

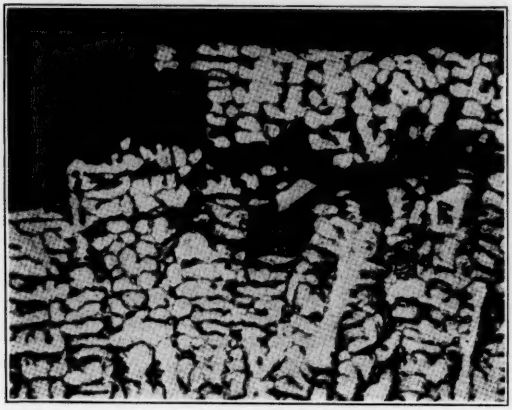


FIGURE 25.—CORRODED BEARING SHOWING LARGE VOIDS WHERE LEAD HAS BEEN REMOVED.

FIGURE 26.—FINE-GRAIN COPPER-LEAD BEARING FROM A HEAVY-DUTY ENGINE WHICH HAS FAILED BECAUSE OF LOW-TEMPERATURE CORROSION—BOTH THE COPPER AND THE LEAD HAVE BEEN REMOVED.

FIGURE 27.—PHOTOMICROGRAPH OF CROSS SECTION OF MAIN BEARING SHELL WHICH IS NOT CORRODED BUT HAS STARTED TO BREAK DOWN BECAUSE OF FATIGUE.

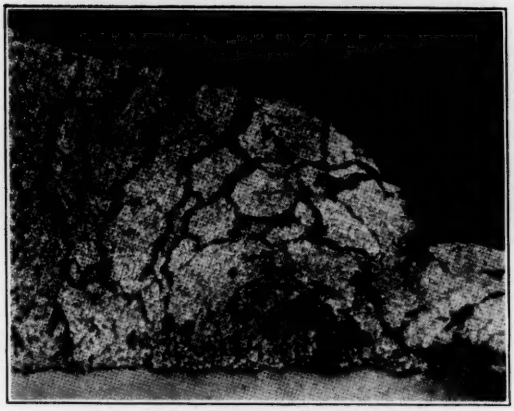


FIGURE 28.—PHOTOMICROGRAPH OF BEARING FROM SAME ENGINE AS THAT OF FIGURE 27 IN WHICH THE FATIGUE HAS PROCEEDED TO TOTAL FAILURE.

For high-duty engines, there is no question that full-length, long jackets should always be used to keep as much heat away from the crankcase as possible, and in view of the considerable increase in heat rejection to the oil in such designs, oil coolers are likely to be a necessity in the future. The heat-interchanger type of cooler in the jacket system is not entirely satisfactory and the probabilities are that the oil coolers should be air-cooled in the same manner as the water radiator is now cooled. Winter conditions, when the cooler would not be needed, can be met by means of thermostat shutters such as are now used on the water radiator, or by by-passing the cooler. Practically all racing engines—the highest-duty of any—regularly use oil coolers.

Another way of keeping copper-lead bearings out of trouble is to pay a good deal of attention to the finish and fit, especially in repair jobs. A copper-lead bearing requires about twice as much clearance as a babbitt bearing, and we have found in many cases that the maintenance forces in fleet operations often do not realize this, and fit replacement bearings as close as babbitt. This will result in a hot-running bearing during break-in, which is likely to sweat the lead out of the mixture, resulting in bearing failure in a relatively short mileage. Moreover, copper-lead bearings wear the crankshaft much more than babbitt does; in some cases the pins and main bearings are trued up and an undersize shell is used, which is then line-reamed to size. If this job is not carefully done, the bearing is practically ruined by the line-reaming. We find that the bearings coming from the manufacturers have a roughness of 3 to 7 microinches, whereas, after line-reaming, in some cases it may be as high as 200 microinches. In addition, the surface of the metal is left in a clean state for immediate corrosive attack. Of course, line-reaming takes off any protective indium coating that may have been applied.

Another important point to consider is the catalytic effect of clean metal. It is quite possible, in an overhauled engine that has been cleaned too much, that the oil will break down very rapidly. Figure 29 shows what

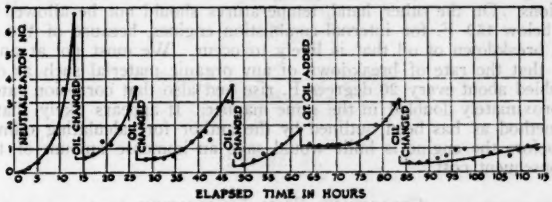


FIGURE 29.—CONDITIONING RUN OF STOCK TEST ENGINE NO. 4, 2000 RPM., 15 HP.

happened in an engine which was completely overhauled, including reboring, new rings, and chemical cleaning. There is a large amount of fresh metal surface which tends to catalyze the oil, and, in addition, during break-in period, particles of colloidal size are coming off the rings and cylinder wall and aggravate this situation considerably. It will be noted that, in a clean engine, the neutralization number of the oil had gone up to nearly 7 in about 14 hours; a change to fresh oil brought it down below 1. The reason, of course, that it did not go down to zero is that some of the dirty oil was left in the crankcase, as is often the condition in many modern designs. In about the same length of time, the neutralization number of the oil had

moved up to 4, and it then is to be noted that the time to reach 4 in the next period was nearly twice as long. In the following periods it is evident that the engine had been conditioned, the catalytic surfaces were poisoned or coated with products of oxidation, and the action was pretty well stopped at the end of 100 hours. It is easy to see, therefore, that overcleaning an engine is not good. The only parts that need cleaning thoroughly are the wearing surfaces. The rest of the engine, provided it is not covered with loose dirt, does not matter and is best left with the coating of varnish or oxidation products that it has acquired.

Deflection or misalignment of as much as 0.002 inch in a 12-inch length can cut the safe load-carrying capacity of a bearing 40 per cent. It is therefore well worth while to see that the alignment is as good as possible between bore and crankshaft, piston pins, and connecting rods. Misalignments caused by distortions of the crankcase in the main bearings are probably the least important, because of length and relative limberness of the crankshaft.

Another trouble, which is not misalignment but distortion, is often found in connecting rods which have been lightened too much. Some of these have quite a limber big end, and it will be found in these designs that the corrosion, when it takes place, is parallel to the crankpin and generally concentrated at the point where the big end would flex the easiest. There are quite a few connecting rods on the market that do flex under normal loads and these are likely to give bearing trouble. It is interesting to note that in racing engines quite heavy connecting rods are always employed. Many have a four-bolt big end.

At the present time it appears that the major trouble with bearings that are showing difficulty in carrying load or corrosion is high temperature. It can now be stated that internal-combustion engines ought to be designed for crankcase temperatures which do not much exceed 180 F. and the lubricating system should be calculated so that there is enough oil flow over all of the bearings to prevent rise of temperature in the bearings themselves from exceeding 35 to 50 degrees. Many engines on the market do not meet these conditions. On the other hand, temperatures should not be allowed to get much below 150 F. for internal-combustion engines, because of the winter sludge breakdown of oil that is likely to occur. We must not at any time forget that the rate of breakdown of any organic material such as oil will be doubled about every 20 degrees F. rise and also that corrosion rates will be approximately doubled in the same manner. It appears likely that some such method as has been outlined by the author for calculating oiling systems before the engine is built would save an immense amount of trouble and consequent cost.

Grease and Solid Lubrication.

Grease lubrication is required for some situations where oil cannot be used; otherwise we probably would not use any grease. A semiplastic material like grease, when in motion, behaves much like a liquid except that it has a variable viscosity with rate of shear. About the only reason for employing grease is to obtain a higher effective viscosity in the bearing than can be obtained with oil, and its use in sleeve bearings is therefore largely confined to bearings running at low surface speed and high load, such as locomotive crankpins, old-style roll-mill necks, and pivot bearings of cranes and turntables. It is also used for cases requiring the ability of the grease to stay in an exposed bearing with much less loss than if oil were used. For this reason grease must be employed for many bearings, such as mine-car journals and exposed conveyer-belt journals, and where hand filling is satisfactory and exposure to weather and water would wash

out an oil. The adhesiveness of grease can be made much higher than that of oil, and the change to a semiplastic solid when the temperature is dropped is one of the reasons for its remaining in bearings without high loss. The best lubricating material is an oil; as lubricants, metallic soaps are not as good, in general, as oils.

At present we have only one solid lubricant of commercial value—finely divided graphite. The experimental data on just how graphite works are limited, the best work having been done in England a few years ago, but practical experience has indicated for a great many years that graphite is a valuable ingredient in greases intended for severe or boundary-lubrication cases and it may also be valuable in oil. Its value in break-in of internal-combustion engines is quite noticeable. The action of graphite appears to be somewhat similar to an oiliness agent or an extreme-pressure additive.

Ball and Roller Bearings.

A word about ball- and roller-bearing lubrication may be of interest since in common practice we often refer to these bearings as "antifriction" bearings. It therefore seems anomalous that bearings which have no friction should need a lubricant. Unfortunately, the beautiful phrase means nothing. Ball bearings do have friction, and at times it may be very considerable. The ball or roller bearing has low starting resistance and the torque is ordinarily not much affected by speed or size for a given load, but if used for high-speed and moderately heavy loads where the friction heat requires a liquid lubricant in quantity to act as a coolant, the friction losses may actually be higher than those for the corresponding sleeve bearing. The friction occuring with a ball and race is made plain from Figure 30.

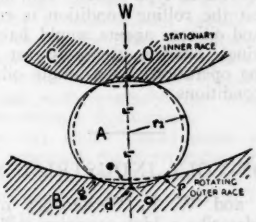


Figure 30.

The ball A under load and the races B and C are indented under the load W. The uncompressed or free shape of the ball and race is shown by dash lines through points d and e. The relative amount of compression of each of the parts is determined by the shape of the race and the modulus of elasticity, and results in an actual contact line at o. There is no slip at o but because the radius r_1 at o is less than the radius r_2 anywhere else on the ball, the surface velocity is slightly greater as we travel from point o to point f or g. Similarly, in the race the compressed-position radius at o is greater than the unloaded-position radius at f; and the surface velocity of the race is slightly lower than at o. As a consequence, there is an increasing slip between A and B, in either direction from o. The same kind of action occurs between A and C, except that the slip is less. Obviously then, heavy load and relatively low surface speed exist, and the conditions indicate the desirability of oiliness agents or greases. Of course, at high

speeds, the surface loads will be kept relatively lighter, but they are in all cases much higher than the surface loads on a sleeve bearing because of very small contact areas. These loads, on large low-speed bearings (for instance those used on roll necks) may exceed 150,000 psi. The rolling friction varies with shape of race for either ball or roller, since the contacts and viscous shears change with the shape.

The second type of loss is largely due to the ball or roller cage, and is also of the viscous-shear type. The cage usually fits quite closely around the ball or rollers. The relative motion gives rise to oil shear just as it does in a sleeve bearing. From this point of view, provided the load requirements under the ball are satisfied, the viscosity should be kept low, since these viscous losses, and consequently the heat generated, increase in proportion to viscosity.

The third type of loss occurs only in a bearing which is flooded with lubricant, or with a grease that flows readily and will not channel, so that it gets in the way of the balls in excess of the slip-lubrication requirements. This loss is of the turbulent type, increasing as the cube of the speed, because the motion of the balls and the cage paddles the lubricant around like a hydraulic brake. It is largely independent of viscosity. It is this third element of friction that gives rise to the high losses in flooded ball bearings at high speed.

The indications are plain. A channeling grease, that will keep out of the way of cage and balls or rollers and only flow a little lubricant as the heat generated melts it down, is desirable to eliminate the turbulent type losses which go up so fast with speed. At low speeds a nonchanneling grease would serve. For lightly loaded high-speed bearings, the oil in the grease should be of as low viscosity as will carry the ball load properly, in order to keep down the viscous-type losses. For heavily loaded, low-speed bearings, it is clear that the rolling condition is controlling, oil of higher viscosity is desirable, and oiliness agents would have distinct value. There is little doubt that a single bearing grease is just as unsuited to all ball-bearing or roller-bearing operations, as a single oil would be for lubricating all sleeve-bearing conditions.

MILLION-VOLT INDUSTRIAL X-RAYS.

Dr. E. E. Charlton and W. F. Westendorp of the General Electric Research Laboratories describe a high-precision million-volt portable X-ray unit in this article from the December, 1941, issue of General Electric Review. Starting with a full knowledge of X-ray generating equipment, an average of six men devoted their full time for four years conducting the research work involved in this project. The transformer, the X-ray tube, and the gas insulation are all radically new. Each contributed its share to the many failures and discouragements and to the difficulty of diagnosing them. This work was started before the advent of the war, but its completion was hastened to meet the industrial needs of the hour.

The new million-volt portable X-ray outfit that is described in this article is the result of further development of the novel features embodied in the million-volt therapeutic X-ray outfit developed and built by the Research Laboratory a few years ago.* In this new equipment, which has successfully been put into production by the General Electric X-Ray Corporation, the

* "A New Million Volt X-ray Outfit," by E. E. Charlton, W. F. Westendorp, L. E. Dempster, and George Hotaling, Jour. App. Physics, vol. 10, p. 374, June 1939.

X-ray tube has been sealed off the pump, and all the component parts of the entire unit have been greatly reduced both in size and weight to make it truly portable.

This was done to increase its flexibility as a radiographic tool for applications in the industrial radiographic examination of metal structures. The radiographic techniques may vary greatly since the materials being inspected cover a wide range in wall thickness, size and shape.

The outfit consists principally of a low-frequency resonance transformer with a coaxially mounted sealed-off multisection X-ray tube within, both contained in a steel tank and insulated with compressed gas. The X-rays are generated from a target mounted in the end of an extension chamber projecting out from one end of the tank.

The attractive features of this type outfit are portability, maneuverability into any position, compactness, simplicity, reliability, freedom from exposed high voltage and ready accessibility of the target. The outfit is three feet in diameter and four feet in length and weighs 1500 pounds.

Transformer.

The transformer shown in Figures 1 and 2 has a low-voltage coil of rectangular wire (part 14 in Figure 1) and a high-voltage coil (9) consisting of 125 thin flat sections spaced apart for cooling.

The resonance principle of operation makes an iron core unnecessary in this type of transformer and the central space of the high-voltage coil is occupied by the X-ray tube, thus facilitating the making of connections to various tube electrodes and providing the benefit of electrostatic shielding by the transformer coil.

The lower end of the high-voltage coil is grounded; the upper or high voltage end is shielded by a rounded and radially slotted brass spinning.

The number of turns chosen for the high-voltage winding is such as to make its natural frequency of oscillation 180 cycles per second. The 180-cycle power is derived from the 60-cycle supply line through the intermediary of a synchronous motor-generator set which also renders operation independent of line-voltage fluctuations.

To avoid the energy loss which would ensue were the magnetic flux of the transformer to penetrate the surrounding steel tank, the latter is provided with an inner lining (11), consisting of narrow overlapping silicon steel strips spotwelded to a thin shell and at the bottom with a ring of similar strips (19).

The multisection high-voltage coil serves a quadruple purpose: to generate voltage; to grade the potential so as to prevent creepage; to shield the X-ray tube electrostatically from the grounded tank; and to support mechanically the rounded top shield. The coil is held under strong compression by seven spring-loaded glass rods, which make it possible for the outfit to be operated in any position.

Continuous operation at 140 F. ambient temperature is made possible by the cooler (1) mounted on top of the tank. A small fan circulates the cooling gas (Freon) through the transformer and over finned water-cooled copper tubing mounted in the cooler. For ambient temperatures below 50 F. electric heaters can be provided to maintain the Freon pressure.

X-Ray Tube.

A twelve-section X-ray tube (Figures 1 and 3) sealed off the pump for operation at one million volts was especially designed to go with this resonance transformer. It has an electron-emitting cathode, a copper-

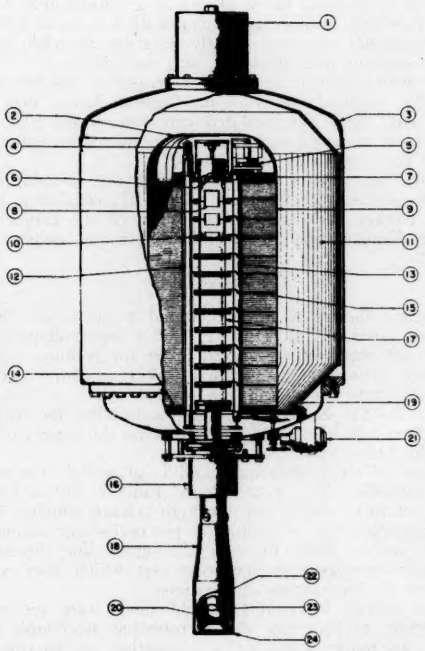


FIGURE 1.—PRINCIPAL ELEMENTS OF MILLION-VOLT PORTABLE X-RAY UNIT.

1: Cooler
2: Slotted brass shield
3: Steel tank
4: Spring for tie rod
5: Variable reactor
6: Cathode assembly
7: End turn filament coil
8: First intermediate electrode
9: Secondary coils
10: Shields around the x-ray tube
11: Laminated shield
12: Glass tie rod
13: Insulating filament-control shaft
14: Primary winding
15: Tap lead
16: Focusing coil
17: Glass envelope
18: Lead shield
19: Laminated steel bottom
20: Tungsten target
21: Filament control motor
22: Water jacket
23: Extension chamber
24: Lead diaphragm

backed tungsten target mounted in the lower end of an extension chamber, and accelerating cylindrical electrodes in each of the intermediate sections.

The tube envelope consists of 12 sections of a molded borosilicate glass tubing joined to fernico rings which carry the intermediate accelerating electrodes of stainless steel. The insulating envelope is 30 inches long and 3½ inches in diameter. The fernico is a special alloy which has an expansion coefficient comparable with one of the borosilicate glasses. This particular property of the fernico alloy, permitting thick metal sections to be fused directly to the glass, has been of major importance in the design of this million-volt X-ray tube. It has allowed the tube to be made much smaller than might otherwise be possible mechanically, and has also permitted a rigorous exhaust of the tube to be carried out before it is sealed off the pump. The inside glass walls of the tube envelope are sandblasted. This procedure greatly increases the voltage that may be applied to each section of the tube without the production of dangerous field current.

The cathode spiral consists of 6½ turns of 8.5-mil tungsten wire mounted in an electrostatic focusing cup. The 11 intermediate accelerating electrodes are made of stainless steel. The target of copper-backed tungsten is mounted in the lower end of the extension chamber, and both target and chamber walls are water-cooled. The tube is supported by a metal flange bolted to the bottom of the grounded metal tank. This flange has been soldered to the copper extension chamber which projects through it.

The tube is sealed off the pump after a specially developed evacuation process has been carried out, during which the glass envelope and metal parts are thoroughly outgassed and all sections are aged at voltages 100 per cent higher than the operating voltage. This process insures stability of operation and long life.

Steel Tank.

The tank of the 1,000,000-volt portable units shown in Figure 4 is made of sheet steel ¼ inch thick, 36 inches in diameter and 48 inches high and has been tested hydrostatically to 200 pounds per square inch. The joint between the shell and bottom flanges is made gastight by means of a rubber gasket.

Gas Insulation.

The insulating gas is derived from a commercial supply tank of liquid dichloro-difluoro methane, CCl_2F_2, known as "Freon-12" or "F-12." The air is first exhausted from the transformer tank to a 28-inch vacuum. The valve of the supply tank temporarily connected is then opened and heat is applied till the pressure in the main tank is about 60 pounds per square inch gage. A pressure-stat on the main tank blocks operation of the transformer if the pressure drops either by leakage or due to abnormally low temperature below 50 pounds per square inch. The gas F-12 is odorless and nontoxic; however, it must never come in contact with an open flame, the resulting decomposition products being poisonous.

Electrical Circuit.

Figure 5 shows a schematic diagram of the principal circuit elements. (The push-button controls, selector switches, control motor drives, interlocks, protective devices, and indicator lights have been omitted for the sake of readability.)

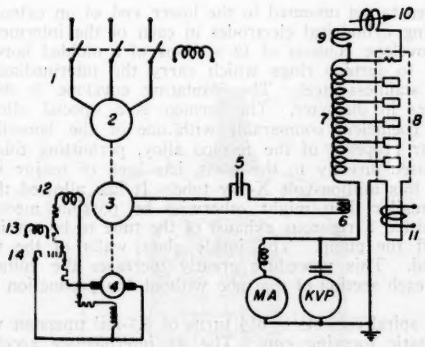


FIGURE 5.—DIAGRAM OF THE PRINCIPAL CIRCUITS OF THE PORTABLE MILLION-VOLT X-RAY OUTFIT.

1: Three-phase 60-cycle line and motor starter
2: Synchronous motor, self-excited
3: Generator for 180 cycles
4: Exciter
5: Series capacitor
6: Primary winding of resonance transformer
7: Secondary winding
8: Million-volt sealed off X-ray tube with auxiliary electrodes
9: End turns for filament power
10: Adjustable reactor
11: Focus coil
12: Generator field
13: Starting relay
14: Kilovoltage adjusting potentiometer

The motor starter (1) connects the 3-phase 60-cycle power line to the motor-generator set. The self-excited synchronous motor (2) drives the 180-cycle generator (3) and the exciter (4). From the generator the 180-cycle power passes through a series capacitor (5) through a long flexible multiple-conductor cable to the primary of the X-ray outfit (6). The series capacitor serves to neutralize the generator reactance and the primary leakage reactance of the transformer, thus loading the generator under all operating conditions with a unity-power-factor load and making the operation independent of the small variations in frequency which occur on power systems. The high-voltage winding (7) is connected through the metering circuit to the grounded extension chamber of the tube (8), through the taps to the intermediate electrodes of the tube and at its upper terminal to the cathode. The filament is operated off end turns (9) on top of the high-voltage coil and the filament current is adjusted by means of a variable reactor (10) driven through a glass insulating shaft by an external filament-control motor.

The metering circuit consists of a d-c milliammeter with a reactor in series and an a-c milliammeter with a capacitor in series. The X-ray tube electron current is a unidirectional current and registers on the d-c milliammeter; the secondary charging current (55 milliamp rms at 1000 Kvp) passes through the capacitor and the a-c milliammeter which is calibrated in kilovolts peak.

AUXILIARY CIRCUIT ELEMENTS.

A selector switch is provided on the control panel (Figure 6) to choose a 0-20 microammeter, a 0-100 microammeter, a 0-1 milliammeter or a 0-5

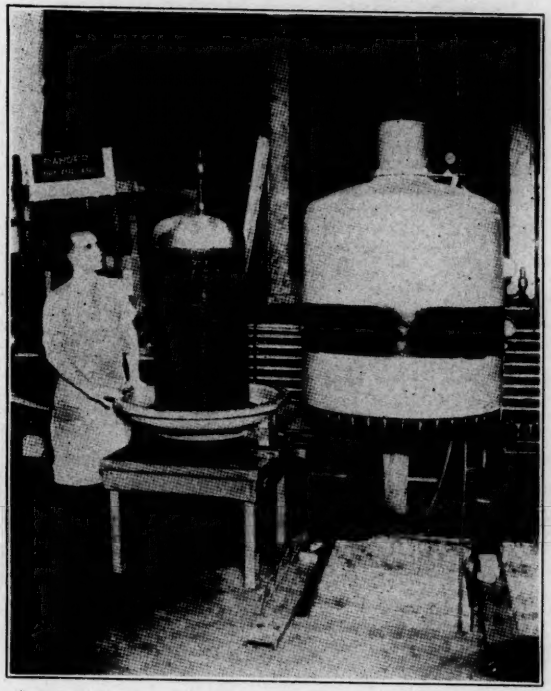


FIGURE 2.—TRANSFORMER REMOVED FROM STEEL TANK OF A MILLION-VOLT X-RAY UNIT.

FIGURE 3.—SEALED-OFF X-RAY TUBE OF A MILLION-VOLT PORTABLE UNIT.

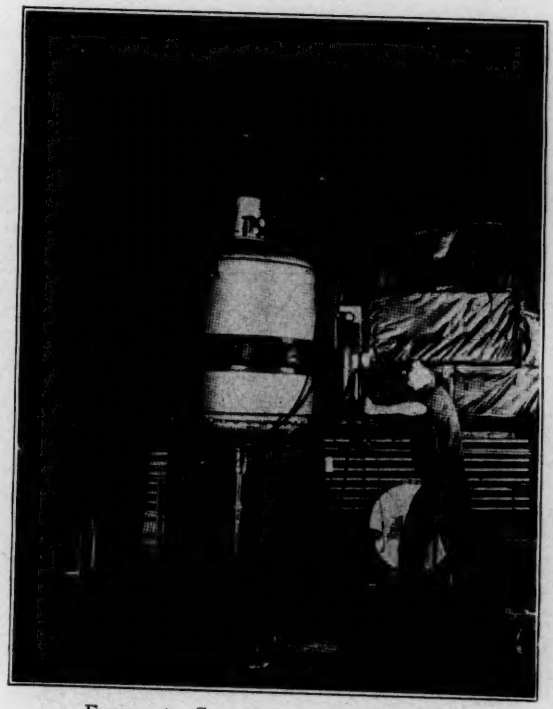


FIGURE 4.—COMPLETELY ASSEMBLED UNIT.

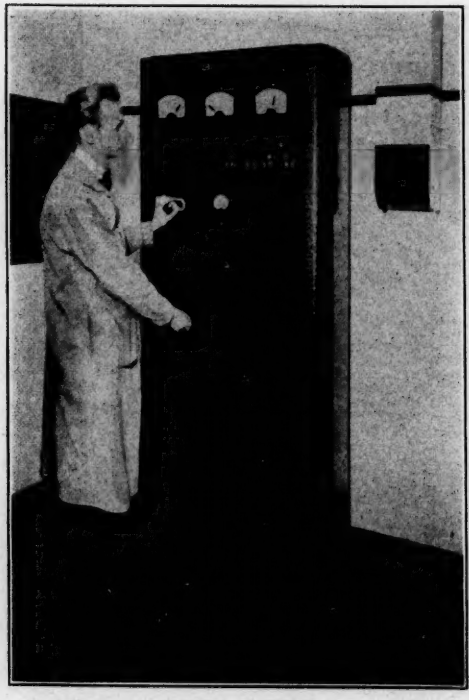


FIGURE 6.—CONTROL PANEL FOR MILLION-VOLT PORTABLE UNIT.

FIGURE 9.—LEAD SHIELD AROUND EXTENSION CHAMBER IMPROVES QUALITY OF RADIOGRAPHS.

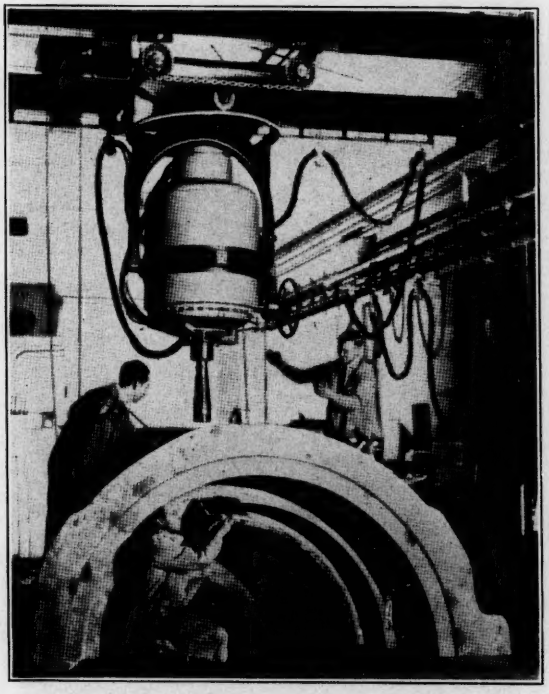


FIGURE 11.—METAL RADIOGRAPHY WITH THE MILLION-VOLT PORTABLE X-RAY UNIT—VERTICAL POSITION.

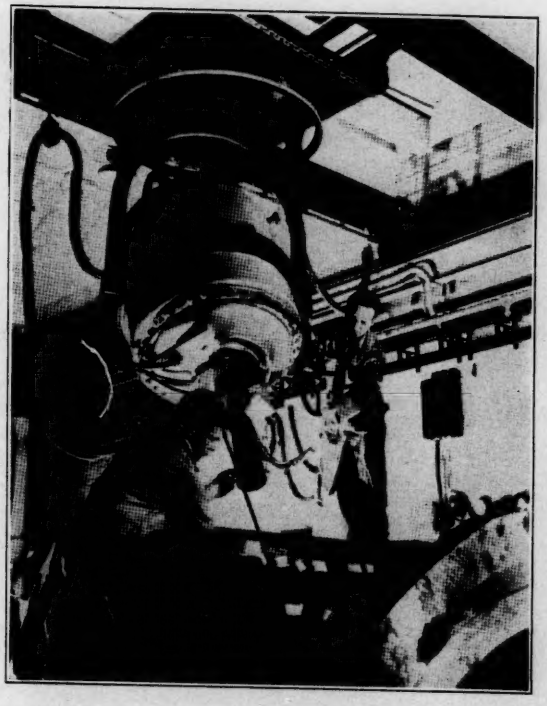


FIGURE 12.—METAL RADIOGRAPHY WITH MILLION-VOLT TRANSMITTED X-RAY BEAM—INCLINED POSITION.

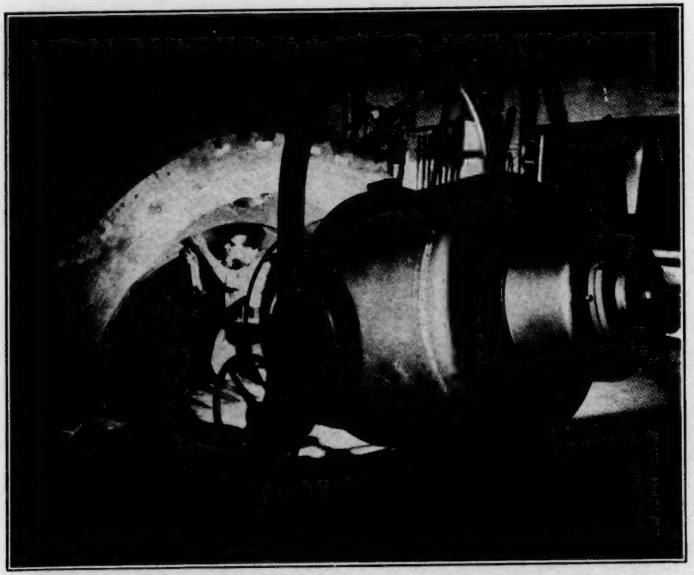


FIGURE 13.—MILLION-VOLT X-RAYS—SETTING UP FOR RADIOGRAPHY OF A THICK METAL SHELL.

milliammeter to read the tube current. The same switch adjusts simultaneously the d-c current through the magnetic focusing coil (11 in Figure 5) surrounding the extension chamber which controls the diameter of the electron beam, thus automatically setting the focal spot size on the target for each of the current ranges. The selector also switches various shunts on a tube current operated relay which protects the outfit and the meters against operation beyond the selected range.

The field (12) of the generator is connected through a push-button operated "X-Ray-On" relay to a motor-driven potentiometer (14) which is connected across the exciter. The exciter can also be used to furnish the direct current for the focusing coil (11). The motor which moves the contact on the potentiometer (14) is operated from the control panel by means of a small "Kvp Control" lever. Interlocking contacts are provided so that the outfit cannot be started unless the potentiometer contact is in its lowest position, corresponding to approximately 500 kilovolts. Another control lever is provided on the panel to operate the filament-control motor which is mounted underneath the tank bottom and geared through a glass shaft to the filament reactor (10) located in the high-voltage terminal cap.

Interlocks are provided for the X-ray-room doors, the cooler motor, the Freon pressure, and the cooling water. There is also relay protection against over-voltage and overcurrent beyond the range selected.

Two timers of 0-5 and 0-30 minute ranges are provided with a selector switch which breaks the interlock circuit and opens a relay (13) after the preadjusted time interval has elapsed.

Operating Sequence.

A typical operating sequence on the control panel of the million-volt portable outfit is instructive. Assume that the operator plans to run his next exposure for one minute at 3.0 milliamp and 1000 Kvp. Before he puts the film in place he adjusts the outfit as follows:

He calls all people out of the X-ray room, closes the doors, selects the 5-milliamp range on the panel, starts the motor-generator set, pushes the "X-Ray-On" button, lets the Kvp meter come up to the starting voltage of 500 Kvp, and then raises the Kvp handle; after he has reached 1000 he adjusts the filament control till he has 3.0 milliamp. He is then ready for the exposure, pushes the "Off" buttons of the X-ray outfit and the motor-generator set and goes into the X-ray room to place his films. For the actual exposure he repeats the foregoing procedure except that he selects the 5-min timer which he has adjusted for the desired one minute of X-ray exposure. After each run the Kvp-control motor returns the potentiometer of the generator field automatically to the starting position.

Design Features.

Transformer.

There are several advantages to be derived from the use of the resonance, rather than the conventional, type of transformer:

(a). The absence of the iron core, permitting the tube to be placed in the axis, conserves the space otherwise required for insulation between core and high-voltage winding and for insulation of the tube. The tank diameter of the resonance transformer is approximately 62 per cent of that of the conventional transformer and the volume about 44 per cent or even less if the space required by the tube is considered. Figure 7 gives a direct comparison of the size and structure of two types.

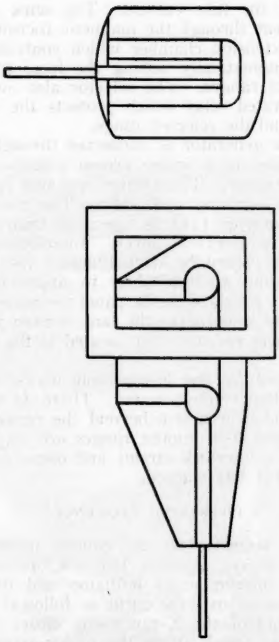


FIGURE 7.—COMPARISON (ON THE SAME SCALE) OF THE RESONANCE-TRANSFORMER DESIGN WITH THE CONVENTIONAL DESIGN OF X-RAY EQUIPMENT.

(b). The waveform in the high-voltage circuit is always sinusoidal regardless of the input voltage. Furthermore, the oscillating current in the high-voltage winding is so large (55 milliamp rms at 1000 Kvp) that the rated half-wave full-load current of 3 milliamp average does not produce any measurable difference between useful and inverse voltages.

(c). In a resonance transformer where the oscillating Kva is 38.5 Kva and the load, including losses, approximately 3.5 Kw maximum, the voltage takes a few cycles for build-up, even if the equipment is switched on suddenly. Therefore switching surges or other disturbances do not raise the output voltage more than a fraction of one per cent, whereas in ordinary transformers 50 per cent or more voltage surges may occur frequently in case improper switching takes place.

(d). The subdivision of the high-voltage winding into 125 thin coils, the bottom 40 spaced ⅛ inch apart, the middle 40 spaced 3/32 inch apart, and the upper coils spaced 1/16 inch apart, results in a very uniform voltage gradient along the length of the stack. Therefore, as indicated in Figure 1, short radial spring taps can be used to supply the proper voltage to all electrodes of the tube.

Since none of the thin flat coil elements sustains more than 10 Kvp and as they have a radial winding depth of several inches, random winding can be used.

For convenience of assembly, start and finish wires of the coil elements are not soldered together but connections are made through flat phosphor-bronze terminal springs as the coils are stacked on top of one another.

Design Constants.

The number of turns in the high-voltage winding is so chosen as to make its natural frequency of oscillation 180 cycles per second. In designing a resonance transformer the standard formula for the natural period of a tuned circuit

$$T = 2\pi \sqrt{LC} \tag{1}$$

may be used, provided that the proper values of inductance and capacitance are chosen to represent the non-uniformly loaded coil and its terminal and distributed capacity. From consideration of the magnetic and electrostatic energy distribution in space, formulas for L and C were derived which allow the designer to predict the required number of coils within two or three per cent.

The inductance is evaluated as

$$L = 25n^2 \frac{D^2}{l_1+l_2+l_3} 10^{-9} \text{ henry} \tag{2}$$

where

n = total number of turns

D = minimum turn diameter plus one third of the difference of minimum and maximum; inches.

l_1 = coil stack height; inches

l_2 = distance from bottom coil to laminated steel disk below it; inches

$l_3 = \frac{1}{4}D$; inches

This latter length l_3 is representative of the distance from the top of the coil to the laminated shell; the shell diameter has been left out of consideration since it bears a close relation to the coil diameter in different designs and has much less effect on the inductance.

The capacitance is

$$C = C_1+C_2+C_3+C_4+1/3\,C_5 \tag{3}$$

C_1, C_2, C_3, and C_4 represent the capacitance of the terminal cap in simplified terms that can be calculated either as parallel plane or concentric cylinder capacitors.

C_1 = parallel plane capacitance of terminal cap top straight up to tank.

C_2 = cylindrical capacitance of cap over to tank wall.

C_3 = parallel plane capacitance of cap bottom through coils to ground, counting the dielectric constant of the gas one, the pressboard coil walls and spacer two, the winding infinite.

C_4 = parallel plane capacitance from cap bottom through inside of coil down to ground, disregarding the effect of the tube electrodes.

C_s = capacitance of the concentric cylinders formed by outside copper turns and inside of laminated shell. This term appears with the coefficient 1/3 in Equation (3) since the actual energy stored in this space is only one third of that with uniform voltage applied.

For parallel planes $C_p = 0.224 \frac{A}{d}$ micro-microfarads, area A and distance d being expressed in square inches and inches. The dielectric constant is assumed to be unity. For concentric cylinders $C_c = 0.613 \frac{h}{\log_{10} D_2/D_1}$ micro-microfarads; h is the height in inches, dielectric constant one.

Thus, for the portable million-volt X-ray transformer is found

$C = 50$ micro-microfarads

$L = 15{,}600$ henrys

For the medical million-volt unit of this type previously developed, the figures are respectively 70 micro-microfarads and 11,150 henrys; and for the oil-immersed 250-Kvp therapeutic X-ray outfit of this type they are respectively 165 micro-microfarads and 4730 henrys.

It is of interest to note that the capacities are of the same order of magnitude as those encountered in radio practice; the enormous inductance, however, brings the natural frequency down to only 180 cycles per second. Three times as many turns would be required for resonance at 60 cycles and the wire diameter necessary for such a design would be impractically small. Higher frequencies than 180 cycles per second could be used but are not desirable since the iron losses in the return magnetic shell and bottom would increase almost proportionally to the square of the frequency even with an increased weight of iron. The proof of this statement can be found in the complete transformation twice per cycle of all electrostatic energy into magnetic energy:

$$\tfrac{1}{2}CV^2 = \tfrac{1}{2}Li^2$$

$$\text{or} \quad \int_v \frac{E^2}{8\pi}\, dv = \int_v \frac{H^2}{8\pi}\, dv$$

The frequency does not enter into these equations. Inside a resonance transformer for 1800 cycles the magnetic field intensity H would be the same as inside a 180-cycle transformer of the same geometry, therefore the flux would be the same, and the iron losses many times greater.

Gas Insulation.

The data as shown by the curve in Figure 8 on the arc-over voltage as a function of the gas pressure were obtained operating the transformer without the X-ray tube. Freon-12 has a dielectric strength 2.5 times that of nitrogen for the same pressure* For an operating voltage of not more than 1000 Kv peak, 50 pounds per square inch gage pressure seems to afford an ample factor of safety. This calls for a weight of F-12 of only about 40 pounds. More than ten thousand pounds of transil oil would be required to give the same insulation in a necessarily larger tank.

* "Dielectric Strength of Insulating Fluids," by Dr. E. E. Charlton and Dr. F. S. Cooper, General Electric Review, vol. 40, p. 438, September 1937.

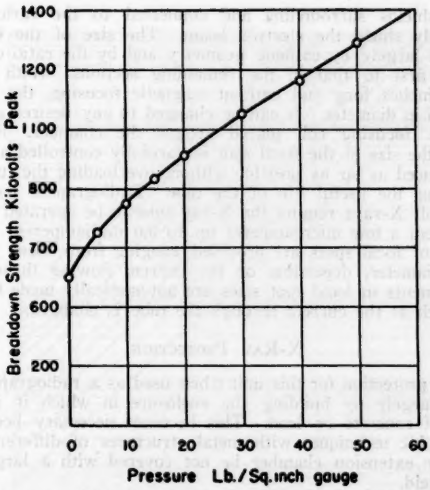


FIGURE 8.—BREAKDOWN STRENGTH OF FREON-12 AS A FUNCTION OF GAS PRESSURE.

Freon-12 decomposes under the action of corona, setting free chlorine and fluorine. Experience with earlier units of this type, employing F-12 as as insulating medium, has already indicated that there is not enough such decomposition taking place under the conditions pertaining in this outfit to cause troublesome deterioration of any of the insulating or other materials present.

TRANSFORMER TIE RODS AND INSULATING SHAFT FOR ADJUSTING CURRENT.

The stack of transformer coils is securely bolted together with seven glass tie rods. These rods, one inch in diameter and 35.5 inches in length, are made of molded hard glass free of air lines and blow holes and with their surface sandblasted to further increase their resistance to surface creepage discharges. The voltage distribution throughout the length of the rods is uniformly graded by placing them close to the transformer coil. They are subjected to a maximum pull of 625 pounds each, by means of a spring in the X-ray unit, and are tested to 1500 pounds to insure an ample safety factor.

The insulating shaft for adjusting the filament current is also made of molded glass and has dimensions and treatment similar to those of the transformer tie rods. This control shaft, mounted inside the transformer stack and driven by a reversible motor, provides remote control for the variable inductance in the filament circuit to give the desired milliamperage.

X-RAY TUBE.

Since the tube is mounted in the axis of the resonance transformer, it is located in a uniform electric field and a weak magnetic field parallel to the tube axis. The magnetic flux does not interfere with electron focusing.

Toroidal shields surrounding and connected to the various electrodes electrostatically shield the electron beam. The size of the electron beam is determined largely by cathode geometry and by the ratio of the voltage used in the first to that in the remaining sections. With an extension chamber 24 inches long and without magnetic focusing, the focal spot is about ⅞ inch in diameter. It can be changed to any desired size by use of the magnetic focusing coil placed around the chamber. For industrial radiography the size of the focal spot is carefully controlled and its dimensions are reduced as far as possible without overloading the tungsten target and shortening the useful life of the tube. Radiographic techniques with the million-volt X-rays require the X-ray tube to be operated over a range of current from a few microamperes up to 3.0 milliamperes. Accordingly, several sizes of focal spots are provided, ranging from about ⅛ inch up to ½ inch in diameter, depending on the current flowing through the tube. These adjustments in focal spot sizes are automatically made by use of the selector switch as the current through the tube is changed.

X-Ray Protection.

The X-ray protection for this unit when used as a radiographic tool must be supplied largely by building the enclosure in which it is used with thick walls of concrete or lead. This becomes necessary because diversified radiographic techniques with metal structures of different shapes demand that the extension chamber be not covered with a large and heavy protective shield.

A lead shield (Figure 9) with one inch wall thickness and weighing 40 pounds is placed around the extension chamber. This does provide some primary protection, but its principal purpose is to improve the quality of the radiographs that may be obtained, by reducing the effects of scattered and stray X-radiation. This lead shield is so designed that it may be quickly adjusted to change the diameter of the cone of radiation of either the reflected or the transmitted beam of X-rays that is being used for a given radiographic procedure.

Radiographic Application.

This X-ray unit, rated to operate continuously at a million volts and a maximum current of 3.0 milliamperes, produces very penetrating X-rays of high intensity. Radiographs of thick sections of steel, ranging from 1 to 8 inches in wall thickness, may be obtained with it in short exposure times. Figure 10 illustrates the exposure times required with million-volt X-rays to take a radiograph through different thicknesses of cast steel at varying target-film distances. Typical action scenes illustrating the radiographic application of the unit in an industrial X-ray laboratory are shown in Figures 11, 12, and 13. Stereoscopic radiographs of metal sections of widely varying wall thickness may be obtained with good definition. This is made possible and practical because of the large amount of penetrating X-radiation that is generated in this unit.

It has been found that radiographs taken with million-volt X-rays will clearly define defects ranging from two per cent down to less than one per cent in steel structures which have walls with thicknesses which vary from 1 to 8 inches. In contrast to industrial radiographic techniques with lower-voltage X-rays, the million-volt X-rays have the added advantage not only of greater radiographic speed but also of allowing greater latitude in the range of metal thicknesses readable on one radiograph without the time-consuming complication of a "blocking" technique as generally practiced in the lower-voltage range.

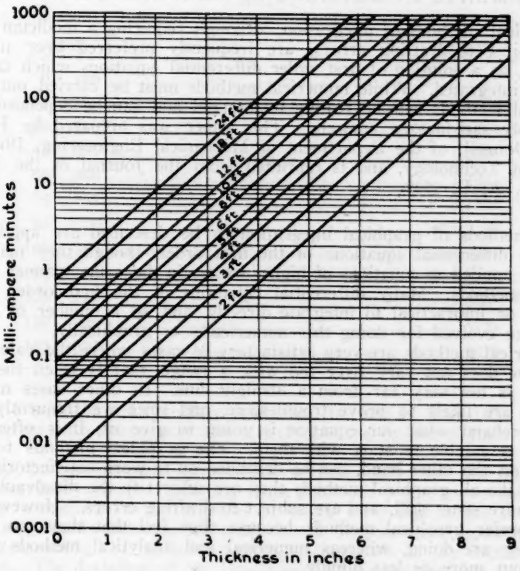


FIGURE 10.—RADIOGRAPHIC EXPOSURE CHART FOR CAST STEEL, USING MILLION-VOLT PORTABLE X-RAY UNIT.

Zero-degree beam—1000 Kvp
0.005-in. lead filter screens
Industrial X-ray film
Film density of 1.0
Family of curves for different target-film distances

The intensity and penetrating power of the beam permits greater target-film distances to be used and still keep the time of exposure down to seconds and minutes. This markedly reduces the distortion of the shadows recorded on the film and also increases the area of the material that can be radiographed with one exposure technique.

CONCLUSION.

This portable million-volt X-ray unit, operating continuously at 3.0 milliamp current, will produce X-rays which both in quantity and quality are equivalent to those obtained from large air-insulated and immovable X-ray units previously designed for medical applications. The ability to move this unit around easily in all directions makes it a flexible and powerful tool for the rapid radiographic examination of castings and metal structures which vary greatly in size and in wall thickness.

REFINEMENTS IN METHODS OF GRAPHICAL INTEGRATION.

Graphical methods of integration, although requiring a modicum of skill and subject to drafting errors, are frequently preferred over numerical methods, as a solution of first order differential equations which cannot be directly integrated. While numerical methods must be carried out blindly, graphical methods can be readily understood and can be depended on for reasonably satisfactory answers. This paper was prepared by Professor L. H. Donnell, of the Department of Mechanical Engineering, Illinois Institute of Technology, and is reprinted from the Journal of the Franklin Institute, April, 1942.

The methods of graphical integration to be described are applicable to ordinary differential equations of the first order, though they may sometimes be applied to equations of higher order, in the same manner as other similar methods. Many differential equations of the first order are impossible or impractical to integrate directly, and so a number of methods have been evolved for doing this numerically or graphically.

Numerical methods are very satisfactory in cases where the value of the derivative does not vary over too wide a range, that is when the plotted function is not very far from a straight line. In other cases numerical methods are likely to prove troublesome, and since we frequently cannot tell beforehand what our equation is going to give us, it is often something of a gamble to start with them. The graphical methods to be described, on the other hand, can be depended on to work satisfactorily in all cases. Like all graphical methods they are subject to the disadvantage that they require some skill, and are subject to drafting errors. However many people prefer graphical methods because they feel that they can see just what they are doing, whereas numerical and analytical methods must be carried out more or less blindly.

As in most methods of graphical integration we start by plotting "equal slope lines." The equations of these lines are obtained by substituting tan $n\theta$ for the derivative in the differential equation. θ is some convenient angle, such as 15 degrees, and n is an integer having values 0, 1, 2, 3 etc. for the different equal slope lines. The quantity tan $n\theta$ is thus a constant for each line (see Table 1) and the equation obtained by substituting it for the derivative is an ordinary algebraic equation, which can usually be plotted very easily. In exceptional cases such equations can always be plotted by using well known methods of cross plotting and interpolation.

As an example, for the differential equation

$$\left(\frac{dy}{dx}\right)^4 - y\frac{dy}{dx} + 3x^3 = 0, \tag{1}$$

the equations of the equal slope lines are obtained by substituting values of tan $n\theta$ for dy/dx. Using $\theta = 15$ degrees and Table 1,

these equations are:

$$\begin{aligned} x &= 0 \quad (n = 0) \\ y &= .019 + 11.2x^3 \quad (n = 1) \\ y &= .193 + 5.2x^3 \quad (n = 2), \text{ etc.} \end{aligned} \tag{2}$$

Each of the equal slope lines is the locus of points at which the integral curve has the *slope* tan $n\theta$, or the *angle* θ. Thus in the above example, any integral curve, that is any graph of the differential equation (1), has a

TABLE 1.

$n\theta$.	$\tan n\theta$.
0°, 180°	0
15°, 195°	.2679
30°, 210°	.5774
45°, 225°	1.000
60°, 240°	1.732
75°, 255°	3.732
90°, 270°	$\infty = 1/0$
105°, 285°	—3.732
120°, 300°	—1.732
135°, 315°	—1.000
150°, 330°	— .5774
165°, 345°	— .2679

zero angle with the x-axis where it crosses the first of lines (2), an angle of 15 degrees with the x-axis where it crosses the second of lines (2), and so on.

Having drawn the equal slope lines, we are ready for the integration. This proceeds by jumps. Starting from a known point, given say by the boundary conditions of the physical problem, we jump to the next equal slope lines; but in between we must resort to guesswork, something that "looks reasonable," or to some approximation such as the methods to be described. The deviation of our line from its true position due to this uncertainty of what to do between the equal slope lines may be small at first, but the error is likely to be cumulative as we go along.

An obvious way to reduce this error is to use more equal slope lines, that is, to use a smaller value for θ. However it does not pay to carry this too far, as the more steps there are in the integration the greater are the unavoidable drafting errors; also the labor involved is more or less proportional to the number of steps. Evidently there is a happy mean somewhere and for most cases a value of 15 degrees for θ will be found satisfactory. With this value all the angles involved can easily be obtained by combining the standard 30 degrees and 45 degrees draftsman's triangles.

It may be noted that where the integral curve has only a slight curvature the intervals between intersections with equal slope lines will be comparatively long, whereas when the curvature is great the intervals will be short. In Figure 1, for example, suppose that the curve abc is the integral curve. The dots represent the points where the curve will intersect the equal slope lines. At a the curve makes an angle of 45 degrees with the x-axis, at the next dot it has an angle of 30 degrees, etc. As the angular changes from dot to dot are equal, the distances between dots—which represent the jumps by which integration would proceed—are great where the curvature is small, near a, and are small where there is a sharp curvature, near b. This is obviously a very advantageous arrangement as it makes the steps of the integration fine where they need to be fine, and coarse where fineness is unnecessary. The usual method of constructing equal slope lines is to plot them from equations obtained by substituting more or less arbitrarily chosen constants for the derivative. This in general results in a wide variation in the fineness of the steps of the integration, with no re-

lationship to the fineness which is desirable at each point. With the method described above, on the other hand, an even degree of accuracy at various parts of the curve is automatically assured.

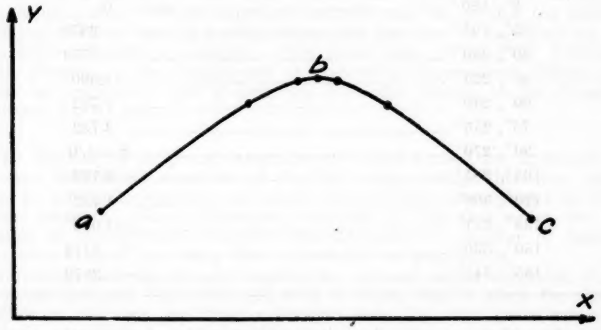


FIGURE 1.

If θ is taken as 15 degrees then there will be only twelve values for tan $n\theta$, as given in Table 1, and hence no more than twelve equal slope lines to plot (unless multiple valued functions, such as anti-trigonometric functions, are involved). In many cases there will be less than twelve lines, as some of the lines will be found to be imaginary, which means that the integral curve never has the corresponding slopes. If very few of the expressions give rise to real lines, this means that the integral curve is nearly straight; in this case one of the numerical methods of integration may be found more satisfactory. In any case only a small section of each equal slope line will be actually used, and some of them may not be used at all, depending on how far it is desired to carry the integration. Hence we can often save much labor if we make a preliminary survey, by going through the whole process in a very rough manner, to get an idea what sections of the equal slope lines need to be drawn in accurately. Or we may draw in the necessary sections of the equal slope lines one at a time, as the integration proceeds and as we need them.

If the equal slope lines happen to be straight lines, it is of course comparatively easy to plot them. Sometimes a transformation of variables can be made in the differential equation which will bring about this result.

Having considered the construction of equal slope lines, let us turn to the drawing of the integral curve, satisfying the differential equation. Two methods are proposed for doing this. The first is very simple and fast and is capable of a high degree of accuracy, but its accuracy depends much on the skill with which it is used. It consists merely in finding a curve, say a section of a French curve, which will cross several of the equal slope lines, each at the proper slope. As guides in doing this, short parallel lines having the proper slope (that is, at angles $n\theta$) are previously drawn through the equal slope lines near the point where it is expected the integral curve will cross. Starting at a known point the integral curve can thus be drawn in steps, quickly and with good accuracy. Figure 2 shows two steps in this process. P is the starting point, given say by the boundary conditions. By trial a curve is found which passes through this point, and is parallel to

the short guide lines where it intersects the nearest equal slope line, and at as many of the succeeding equal slope lines as possible. After the first section is drawn in, the integral curve is extended another short distance in the same manner, and so on. If the short guide lines are drawn only as needed, only a few of them will be required.

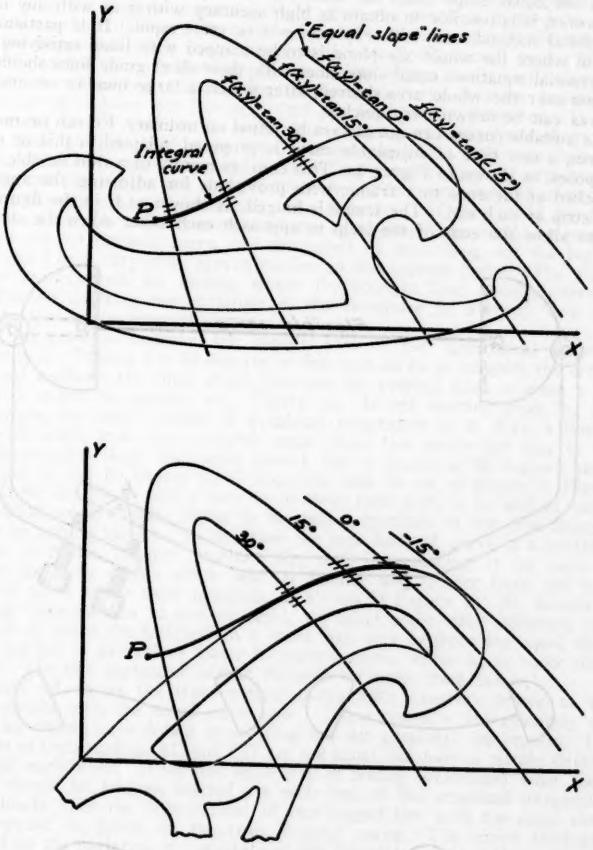


FIGURE 2.

For the greatest accuracy, curves should be found which fit at as many points at once as possible—in any case at not less than three points. Greater accuracy can also be secured by using only the middle portion of

the curve which fits. Thus we may fill in only one space at a time—between two adjacent equal slope lines—but for each step we attempt to find a curve which fits not only at these two equal slope lines but also at the two adjacent equal slope lines.

This method requires a good eye, not only for making sure that the selected curve is parallel to the short guide lines at the point of intersection with an equal slope line, but also for quickly locating suitable curves. However, it is possible to obtain as high accuracy with it as with any other graphical method, and with experience it is very rapid. It is particularly useful where the whole x,y plane is to be mapped with lines satisfying the differential equation; equal slope lines with their short guide lines should be drawn over the whole area desired, after which a large number of integral curves can be drawn very quickly.

As suitable curves can not always be found on ordinary French or marine curves, a new type of adjustable curve is proposed, adapted to this or other purposes, as shown in Figure 3. This consists merely of a thin flexible strip attached at the ends to a frame, with provisions for adjusting the angle of the strip at each end. The frame is hinged, as shown at h in the figure, so as to allow the ends of the strip to approach each other when the strip is

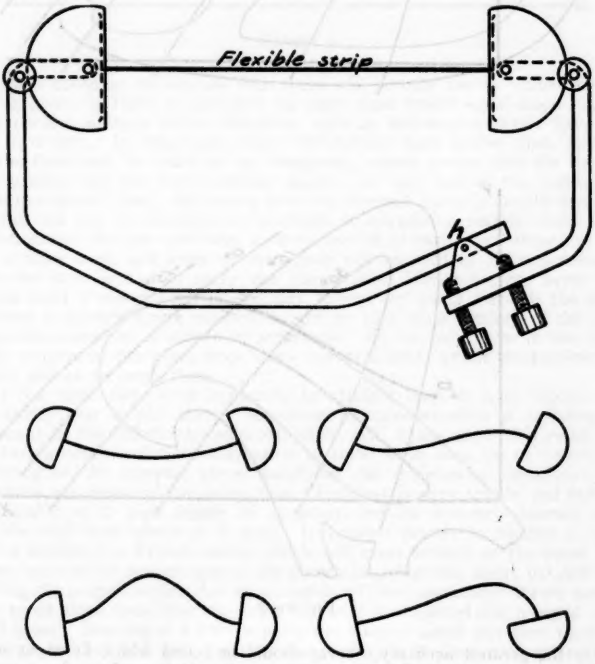


FIGURE 3.

bent. By suitably adjusting the angles of the ends of the strip a very wide range of curves can be secured, as suggested in the figure, and in the photographs of a model. An adjustment can also be provided at the hinge, as shown, to allow longitudinal loads to be applied to the strip, which somewhat increases the range of curves which can be secured. For small scale work the strip may be of spring brass about 3 inches long, .005 inch thick and .5 inch high. In use, the strip should be pressed tightly against the paper with the fingers of the left hand when the pencil is applied, to prevent its moving. Or the strip may be beveled to a sharp edge at the bottom, and the impression made in the paper used instead of a pencil mark; if the paper is backed up with blotting paper a legible impression is easily secured.

The second method for drawing the integral curve is more automatic, requiring more labor but less skill than the method just described. It is a refinement of the well known method of drawing straight lines of proper slope through the equal slope lines, intersecting about half way between them, and then smoothing out the resulting jagged line. It consists in finding out *just where* the straight lines *should* intersect, instead of merely letting them intersect "about half way between." If this could be done exactly, the points of intersection with the equal slope lines would be exactly on the correct integral curve, and the result of smoothing out the jagged line would be a very close approximation to this correct curve. The method to be described, for finding where the straight lines should intersect, depends on studying the variation in the curvature of a rough first approximation to the integral curve.

As an example consider the simple case of the differential equation $dy/dx = x$. Taking $\theta = 30$ degrees in this case, so as to magnify the errors in usual methods, the equal slope lines are the vertical lines $x = \tan 0$ degrees, $x = \tan 30$ degrees, etc., Figure 4a. If our starting point is, say, the origin, the usual method of graphical integration is to draw a line at 0 degree angle from the 0 degree equal slope line about half way to the 30 degree equal slope line; from here a line is drawn at 30 degrees about half way to the 60 degree equal slope line, and so on, as shown in Figure 4b. If we do this in such a way as to make lines such as bc and cd equal, then what we are really doing is to replace portions of the true integral curve by arcs of circles. In this case the true integral curve is a parabola whose curvature changes continuously, and in replacing it by arcs of circles we make errors which lead us farther and farther from the true integral curve as the work proceeds, as shown in Figure 4c. If, instead of making lines such as bc and cd equal, we adopt some other arbitrary rule such as to make the intersection c come half way between the equal slope lines, we are in general as likely to make matters worse as to make them better. (In this particular case it happens to make them better.)

Figure 4d shows the true integral curve with tangents drawn at the intersections with the equal slope lines. These tangents are evidently the lines we should have drawn in carrying out our graphical integration. The lengths of lines such as BC and CD are not equal, but have a certain definite ratio to each other. If we had some way of telling beforehand what these ratios should be, and we carried out each step of the graphical integration accordingly, then the intersections of our jagged line with the equal slope lines would be points on the true integral curve. The ratios obviously depend on the variation in curvature of the integral curve; if the curvature were uniform BC would equal CD; if, as in the present case, the curvature decreases from B to D, then BC will be shorter than CD, and vice versa.

The first step in the method is to construct a rough first approximation to the integral curve, so that we may study its variation in curvature. This approximate curve may be drawn by any method—even a curve drawn

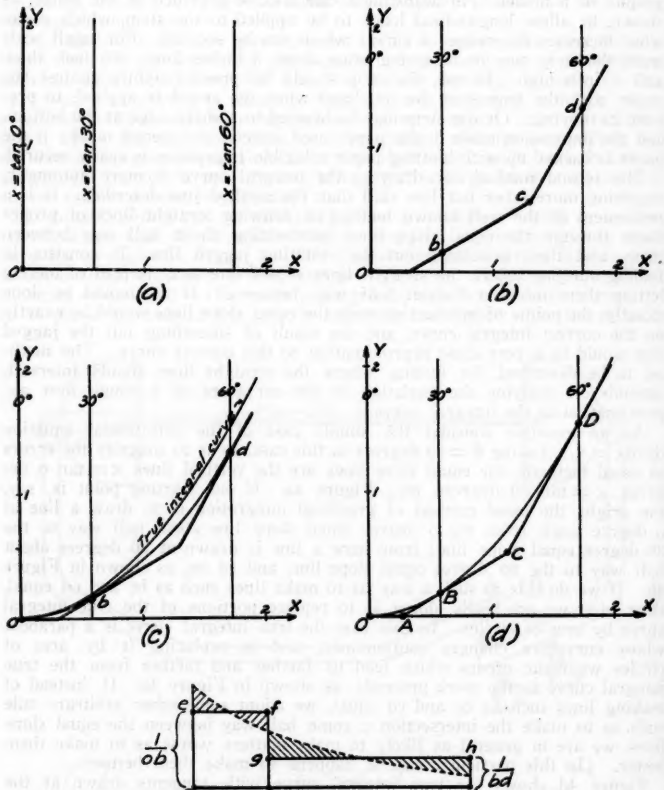

Y
x = tan 0°
x = tan 30°
x = tan 60°
(a)
0°
30°
60°
(b)
True integral curve
(c)
A
B
C
D
(d)
(e)
(f)
c.g.
OA approx.
BC approx.

FIGURE 4.

free-hand, so as to have roughly the proper slopes where it crosses the equal slope lines, may be sufficiently accurate. The next step is to make a development of this approximate integral curve, showing the points of intersection with the equal slope lines. Now the average curvature of the integral curve, between intersections with two equal slope lines, is inversely proportional to its length between these intersections; this is easily seen if we remember that the total angular change between these intersections is a constant, ϑ. Hence if, on each interval of the development, we plot the reciprocal of the length of this interval, we obtain a "curvature graph," that is a graph depicting the variation in curvature along the integral curve.

Going back to our example, suppose we took as our first approximation to the integral curve the curve obd, Figure 4c. The development of this is the line obd in Figure 4e, where o, b and d are the intersections with equal slope lines. Measuring the lengths of the intervals ob and bd to any scale, calculating the reciprocals of these lengths, and plotting these reciprocals on the corresponding intervals, we obtain the line efgh. This line represents graphically the variation in curvature along the original curve obd, Figure 4c, because the average curvature from o to d is ϑ/ob and the average curvature from b to d is ϑ/bd.

The curvature of the *true* integral curve will have nearly the same average value between equal slope lines as our first approximation, but it will vary continuously instead of in jumps. A close estimate of the variation in curvature of the true integral curve is therefore obtained by smoothing up the graph efgh, as indicated by the dotted line in Figure 4e or the line ijk in Figure 4f. In doing this we should try to keep the *areas* for each interval the same as before, by making the similarly shaded areas in the figure approximately equal.

Having obtained the "curvature graph" ijk, showing to some scale (the scale does not matter) the variation of curvature along the integral curve, the desired ratio between the lengths of lines such as BC and CD is very simply obtained. It is equal to the ratio between the distances from the center of gravity of the area bjkd to the ends jb and kd; or in other words, the c.g. of the area bjkd divides the interval bd in the desired ratio of BC to CD. In practice the total lengths bd and BCD are usually nearly the same, especially when a small value of ϑ has been used. In such a case it is sufficiently accurate to take the distances of the c.g.'s from the left hand ends of the corresponding intervals as equal to the lengths OA, BC etc., as indicated in Figure 4f. Hence as soon as we have determined the curvature graph, such as ijk, the construction of the second approximation to the integral curve is easy; we merely lay off at a 0 degree angle the length OA as determined from the curvature graph; from A we draw ABC at 30 degrees, taking the length BC from the same graph, and so on.

If there is considerable difference between lengths such as bd and BCD, then we must carry out each step of the construction in such a way as to make the ratio of lines such as BC to CD equal to the ratio in which bd is divided by the c.g. of the area above it. This can be done with sufficient accuracy by marking the dimensions from the curvature graph on the edge of a sheet of paper and comparing them with tentative lines on the other diagram, until, by eye, the same ratio is obtained.

The determination of the c.g.'s of areas such as oijb and bjkd can most easily be carried out by changing the areas into equivalent trapezoids, as shown in Figure 5, being careful to make the positive and negative shaded areas roughly balanced about the center, so that the location of the center of gravity will not be seriously affected. The center of gravity of the trapezoid may then be determined by well known formulas or graphical methods, as indicated in the figure.

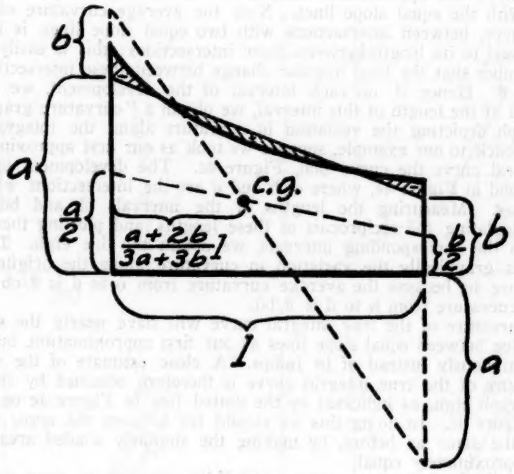


FIGURE 5.

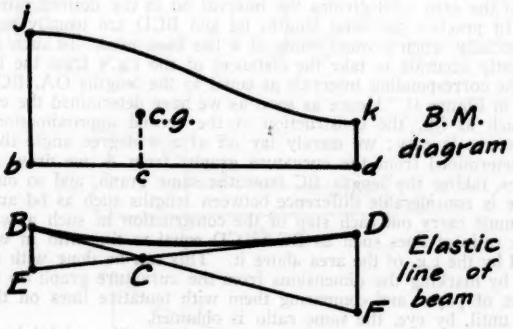


FIGURE 6.

The proof that the c.g. of areas such as bjkd divides the interval bd in the same ratio as BC to CD is most easily given by using an analogy with the well known Area Moments method of studying the deflection of beams. If, in Figure 6, the curve BD represents the elastic line of a bent beam, with tangents BCF and DCE drawn at its ends, and bjkd is its bending moment diagram, having the c.g. shown and the area A, then by the Area Moment laws:

$$BE = (A/EI)\,bc, \qquad DF = (A/EI)\ \ cd. \tag{3}$$

Hence $bc/cd = BE/DF = BC/CD$ by similar triangles. The relation between an integral curve and its "curvature graph" is exactly the same as exists between a bent beam and its bending moment diagram, as the curvature of a beam is everywhere proportional to the bending moment. Hence the above proof applies to the similarly lettered lines in Figures 4d and 4f. The proof is, of course, strict only if the angle between BC and CD is small, but the error is slight when this angle is 15 degrees or even 30 degrees.

A less crude example of the application of this method is given in Figures 7 and 8. Again θ is taken rather large to magnify the chance for error. The figures deal with the same problem in the same way, the difference being that in Figure 7 θ is taken as 22.5 degrees and in Figure 8 as 30 degrees. The differential equation is $dy/dx = \sin\ (x + y)$, with the

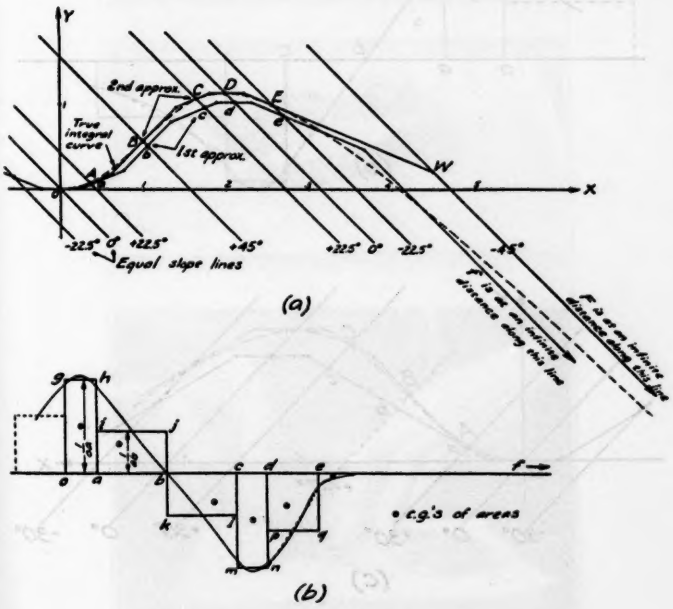


FIGURE 7.

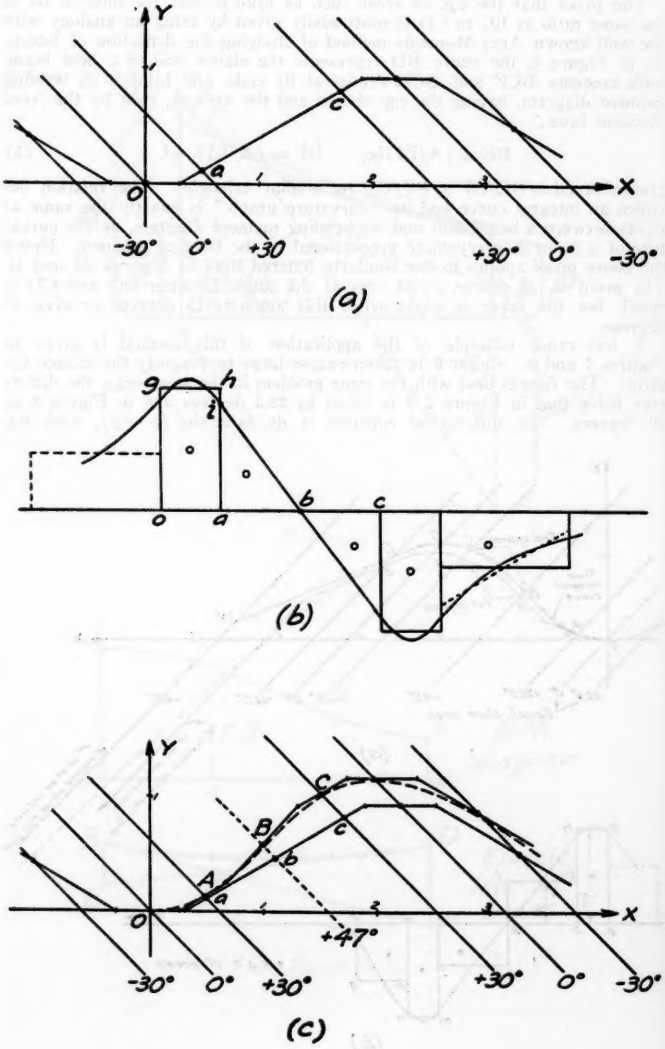
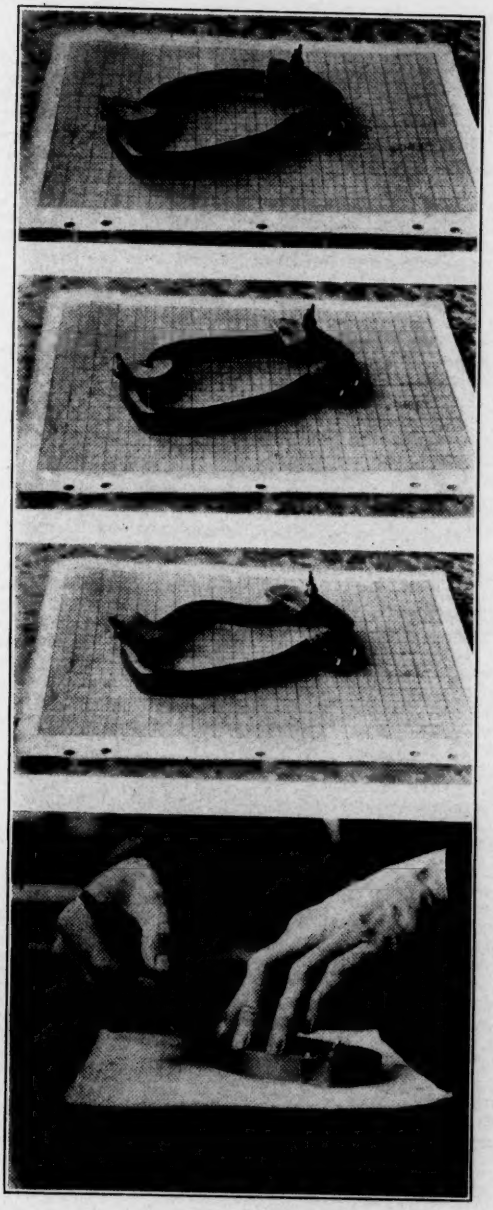


FIGURE 8.



origin as the starting point. This equation can also be integrated mathematically, and of course in practice there is seldom any point to using graphical integration in such a case. However it is better for an illustration, as we have an exact solution, shown in broken lines in the figures, with which to compare our method. This problem is a difficult test of any method, as the true integral curve has a point of inflection and approaches one of the equal slope lines asymptotically.

The rough first approximation is obtained by the method of drawing lines of proper slope through the equal slope lines and intersecting half way between them. This was chosen in order to compare the inaccuracies involved in this method with the refinement possible with the second approximation, and also because it is probably the simplest and most nearly "automatic" of all methods of graphical integration.

In Figure 7a the points a, b, c, d, e, and f are the intersections with the equal slope lines obtained by the first approximation. The development of this first approximation is given in Figure 7b. It makes little difference whether a development is made of the jagged line or of a smooth curve drawn through the points a, b, c, etc. It is of course much easier to develop the jagged line, as was done in this case; the points were simply marked in order along the edge of a sheet of paper held successively against each portion of the jagged line in Figure 7a, and then transferred to Figure 7b; this is much easier than using dividers.

Next, the lengths of the intervals oa, ab, etc., Figure 7b, were measured according to a convenient scale, their reciprocals were calculated and plotted on the corresponding intervals, giving us the jagged "curvature graph" g h i j k l, etc. The reciprocals are plotted in the positive or negative directions according as the curvature in the corresponding section of the integral curve, Figure 7a, is concave upward or downward. There need be no confusion about this, as it makes no difference which convention is used for positive and negative curvatures, and it is always perfectly obvious when the curvature changes sign.

The jagged curvature graph was then smoothed up as shown, care being taken to keep the areas over each interval roughly the same. The center of gravity of the area over each interval was figured by the formula given in Figure 6 and the second approximation to the integral curve was drawn accordingly, as previously described, giving us the points A, B, C, D, E and F in Figure 7a. It will be noted that according to the construction, since the length ef is found to be infinite, the ratio of EW to WF is given as zero; hence the point F, the intersection of the integral curve with the —45 degree equal slope line, is an infinite distance from W along this line, as it should be, since the exact curve approaches the line asymptotically.

Figure 8, in which θ is taken as 30 degrees, illustrates a case where the method as so far described apparently fails. However if the meaning of the various steps in the process is kept in mind, then it is always quite evident in such cases what must be done. In Figure 7 it happened that one of the equal slope lines passed through the point of inflection of the integral curve and this saved us from trouble. In Figure 8a we are not so lucky. The equal slope lines at the ends of the interval ac are both +30 degrees. Since the total angular change of the integral curve over this interval is zero, its average curvature must be zero. Hence in constructing the curvature graph, Figure 8b, the ordinate for the interval ac should be zero, instead of 1/ac as ordinarily. There is no particular danger of making a mistake and plotting the reciprocal in such a case, because it must be decided whether to plot the ordinate in the positive or negative direction, and as soon as this question comes up it at once becomes evi-

dent that the curvature is neither positive nor negative, and hence the ordinate must be zero.

Another difficulty presents itself when we attempt to find the center of gravity of the area on ac, Figure 8b. As the area is composed of equal positive and negative parts its center of gravity is an infinite distance away, and the construction of the second integral curve at this point becomes indeterminate. The obvious way around this difficulty is to insert another equal slope line at the point of inflection, as shown by the dotted line in Figure 8c. The point where the curvature graph crosses the development of the first integral curve, b in Figure 8b, is plotted back onto the first integral curve, and the new equal slope line is drawn through this point, parallel or approximately parallel to the neighboring equal slope lines. The angle corresponding to this new equal slope line can readily be found from the curvature graph. The *area* under this graph between two points represents the *angular* change in the integral curve between these points. Hence we can find the angle corresponding to the new equal slope line from the area of the curvature graph between it and a neighboring equal slope line. In this case it is found by trial that the area aib, Figure 8b, is about 17/30 of the area of any of the rectangles in the figure, each of which represents an angular change of 30 degrees. The area aib therefore represents an angular change of about 17 degrees, and the angle corresponding to the new equal slope line through b is given as 47 degrees (17 degrees added to the angle at a). Having determined this new equal slope line, the work can continue as before.

If, in Figure 7, the points determined for the second approximation to the integral curve, A, B, C etc., are compared with the exact integral curve shown in broken lines, it will be seen that the errors in this second approximation are very small, in spite of the fact that θ was taken as large as 22.5 degrees and the work was done with no more than ordinary care. The points do not lie exactly on the true line, of course, but the errors were of the order of magnitude of the width of the pencil lines used in the construction, and were therefore probably due as much to drafting errors as to errors inherent in the method. A third approximation can of course be made by constructing a development of the second approximation and so on, but experience indicates that generally this is unnecessary, and is useless unless the work is done with extreme care. The labor of carrying out this method is not very great—in fact it is much easier to do the work than to describe it. No more than ordinary drafting skill is required, and good results can be obtained even if the work on the curvature graph is done very carelessly or approximately—that is, better results are obtained by using the method roughly than by using less exact methods carefully.

WELDING JIGS AND MANIPULATORS.

This abridged article describes how the use of jigs and manipulators can speed up welding production as much as one hundred per cent while simultaneously reducing the skill required of the welder. It is taken from Welding Memorandum Number 3, published by the Advisory Service on Welding, Ministry of Supply, London. The abstract is reprinted from Engineering, November 28, 1941.

The term "jig" is used for an appliance to hold individual parts of a component together in correct relationship for welding. The term "manipulator" is used for an appliance by which assemblies to be welded are

brought quickly into the most suitable positions with the minimum of waste effort on the part of the welding operator. In general, the welding operating factor is about 25 per cent, that is, the average welding operator is depositing weld metal about 25 per cent of his working time, the remainder being spent in assembling, tack welding, turning the work by crane or by man-handling, changing electrodes and cleaning welds. The use of jigs and manipulators can obviate much of this waste and ensure that the operator is depositing weld metal for a higher percentage of his working time. Shops using adequate jigs and manipulators have increased the welding factor to 50 per cent.

Jigs may be used to perform one or more of the following functions:—(1) to hold parts in correct relationship to one another for welding, ensuring uniformity of size, shape and position of components, and avoiding the necessity for tack welds; (2) to control distortion due to the heat effect on the work, by forcible restraint and by dissipating the heat in the jig structure; (3) to facilitate quick and accurate assembly of components; and (4) to reduce the need for highly skilled operators on repetition work.

The simplest form of jig is the clamping device shown in Figure 5. Clamps of this type have the advantage that any thickness of spacing block can be used to suit the work to be clamped, but the operator has three loose pieces to handle—the work, the clamp bar and the space block. The number of parts to be handled can be reduced by welding the spacing block to the clamp bar, or by making an L-shaped clamp bar, but this necessitates a number of clamp bars suitable for work of different thicknesses. Careful planning of shop production can reduce this difficulty by arranging that each group of welders always handles the same class of work. Where space permits, a light spring can be placed under the clamp bar (see Figure 6) which holds the bar away from the bedplate while the work is being assembled.

Where the design of the work is such that there are spaces between components, it is frequently possible to arrange clamp bars to hold two details simultaneously. Such a clamp should be placed so that it is not necessary to remove the nut and clamp bar to release the work, but a slackening of the nut and winging the clamp bar through 90 degrees will effect the release. Here again, a spring under the clamp bar will greatly facilitate handling. A threaded handle made from a bar welded to a nut may be substituted for the conventional nut, thereby eliminating the need for a spanner.

Quick-acting toggle or cam clamps effect a considerable saving in time. Figures 7 and 8 illustrate a simple form of toggle clamp in the closed and open position. The pressure required on the work is adjusted in the first instance by the adjusting screw (which, in Figures 7 and 8, is shown through a tapped hole in the clamp arm), and locked with a lock nut. When clamping an awkward section, a toggle clamp as shown in Figure 9 can be advantageously used. Adjustment is made by the two nuts shown above and below the toggle bar, because the shape of the pressure pad prevents adjustment by turning the adjusting bar itself. Any form of toggle clamp must be robust and adequate. The mechanism must bend to some extent, and the final pressure is not equal to the maximum pressure. This point should be watched if the clamps have to offer restraint to distortion.

For clamping plates together, a V-clamp (Figure 11), cut from a plate or bar, can be hammered on to the work; or else a C-clamp (Figure 10) is used and adjusted by means of a bolt. A much quicker method is by using the form of toggle clamp shown in Figure 12.

When a large number of similar products is being assembled, it should prove an economy to use a combination of toggle clamps and vice, similar

to that illustrated in Figure 1. The toggle clamps are coupled, and are operated simultaneously by the hand lever. The vice is operated by the handwheel. A vice was chosen to give a very heavy and rigid clamping pressure. The vice jaws are pivoted to allow for slight irregularities in the work. The clamp jaws are easily removable and different sizes of jaws can be used for varying sizes of work.

The welding of longitudinal seams of containers, tubes and similar products can be facilitated by the use of a scissor type of clamp. For short lengths in thin sheet metal, a hand-operated clamp, as shown in Figure 13 is suitable. The sectional view shows the grooved backing bar. When the clamp is in use, the groove lies underneath the weld seam and helps to ensure complete penetration. The clamping fixture shown in Figure 14 was designed for longitudinal seams on heavy material. Two top clamp bars are fitted, and the backing bar is arranged on a swinging beam to facilitate unloading. In the particular design shown, the backing bar is made in sections and clamping is effected by forcing these sections against the fixed top beams by air pressure in the rubber hose, as illustrated in Figure 14. Figure 2 shows a jig with 150 details assembled for welding. Note that the entire working table of this jig can be tilted; thus it is a combined jig and manipulator.

Jigs for welded work may be designed either for the mass production of one article or may be adaptable for a variety of articles. In designing them, consideration must be given to the following points:—(a) freedom of access to all weld joints; (b) effect of welding heat upon the locating members of the jig and upon the component parts of the product being welded; (c) ease of loading and unloading the jig, which has a pronounced effect upon the daily output of welding per operator; (d) the jig should be so designed that the component parts of the welded product can only be inserted in the correct position, to enable semi-skilled labor to be employed in loading the jig; (e) clamping elements should be arranged so that spatter from the welding does not fall upon threaded parts or machined faces; (f) clamping elements must be arranged so that distortion is controlled; (g) clamping elements subject to wear should be made of a hard-wearing alloy or have a wear-resisting alloy deposited by welding; (h) assembly of the component parts to be welded should be arranged, so that tack welding is not necessary; (i) it should not be necessary to remove the article from the jig until all welding is completed; (j) the jig should be designed so that distortion of the component parts during welding does not lock the article in the jig and hinder removal from the jig when welding is completed; and (k) if the jig is not designed as a manipulator (i.e., if it cannot be rotated) it should be light in weight to permit easy change of welding position. The behavior of work in a welding jig is not always predictable, and, in designing and manufacturing a jig, allowance should be made for possible alterations which may be necessary.

Whenever possible, work should be mounted on a device by means of which the welding operator can readily rotate or tilt it into the most convenient position for flat downhand welding. By means of such equipment, a higher daily output of weld per operator is secured. Welding a fillet, for instance, in the tilted position, instead of in the horizontal-vertical position, means that the weld can be completed with, say, one run of a four-gauge electrode instead of three runs with an eight-gauge electrode; that is, in less than half the time, since two descaling operations are eliminated, with less danger of slag inclusion in the weld and less danger of undercut.

An obvious use for turning equipment is in welding the circumferential seams of cylindrical objects, flanges on to pipes, etc. Rollers for this work can be adjusted to accommodate different sizes of job, as in Figure 15,

FIGURE 1.

FIGURE 2.

FIGURE 3.

FIGURE 4.

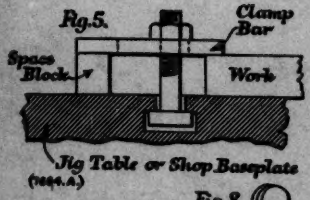


Fig.5.
Clamp Bar
Space Block
Work
Jig Table or Shop Baseplate
(1884.A)

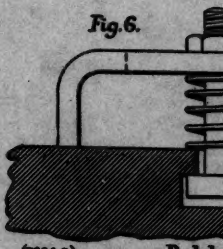


Fig.6.
(1884.B)

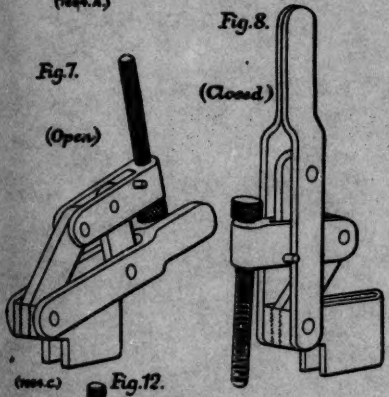


Fig.7.
(Open)
Fig.8.
(Closed)
(1884.C)

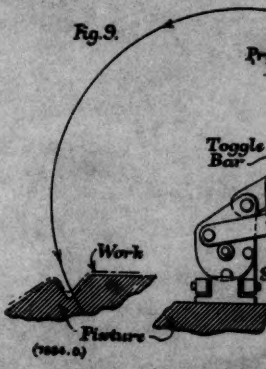


Fig.9.
Toggle Bar
Work
Fixture
(1884.D)

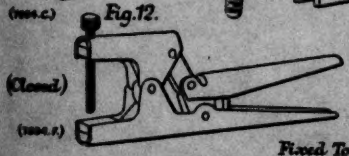


Fig.12.
(Closed)
(1884.E)

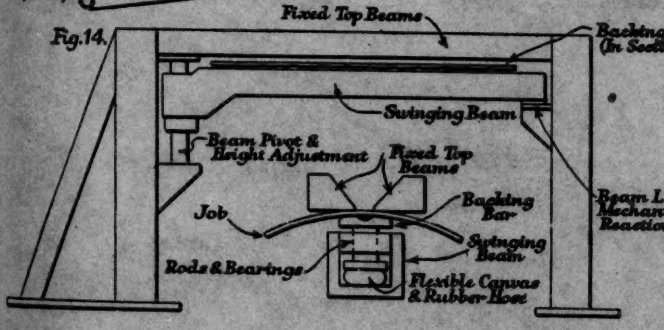


Fig.14.
Fixed Top Beams
Swinging Beam
Beam Pivot & Height Adjustment
Fixed Top Beams
Backing Bar
Job
Swinging Beam
Rods & Bearings
Flexible Canvas & Rubber Hose

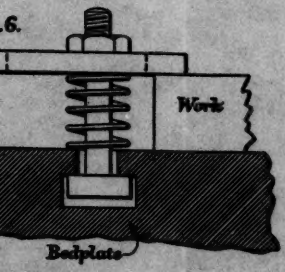


Fig.10.

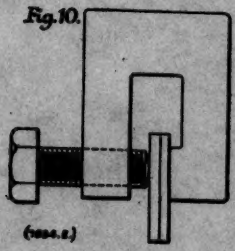
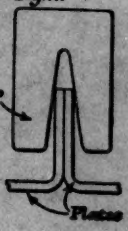
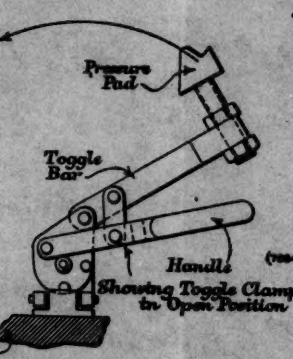


Fig.11.





SECTION

Fig.13.

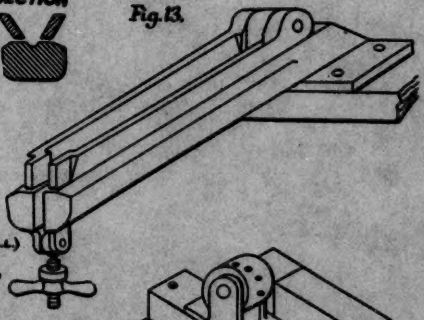
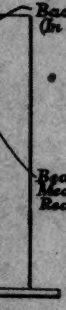
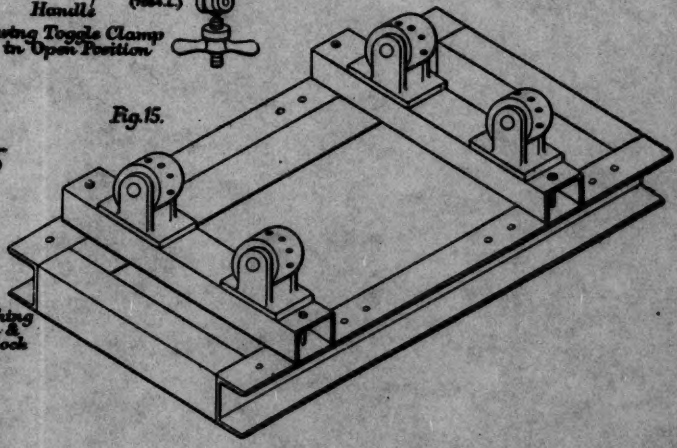


Fig.15.





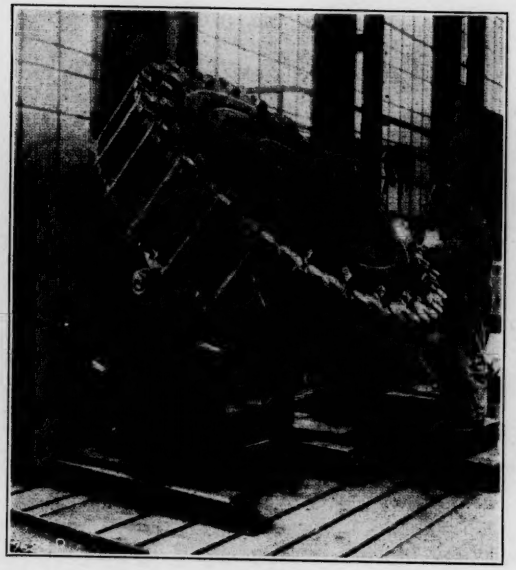


FIGURE 16.

FIGURE 17.

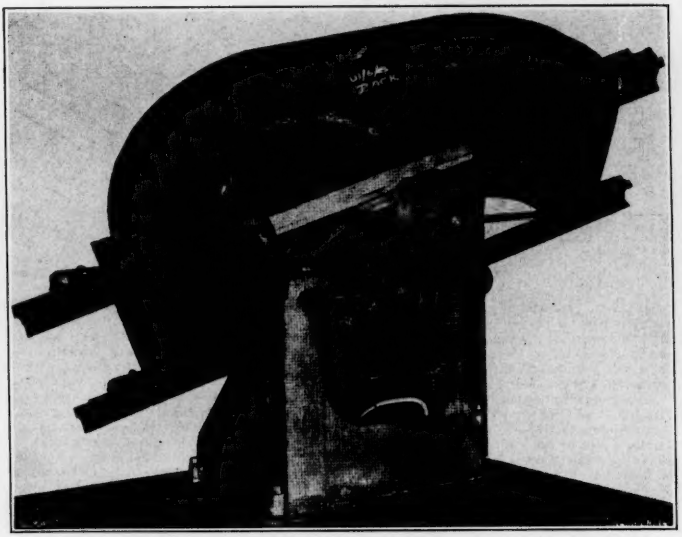


FIGURE 18.

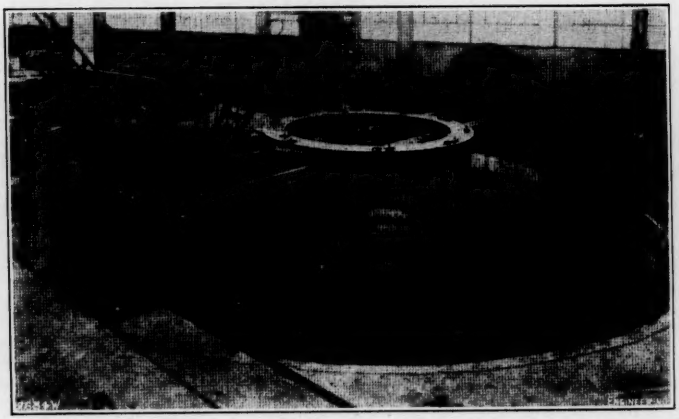


FIGURE 22.

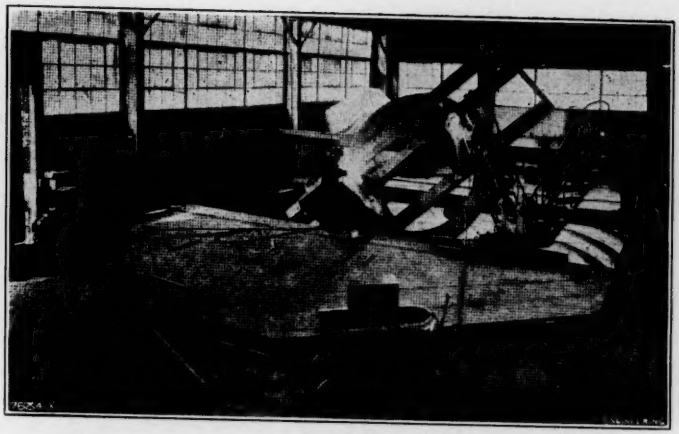


FIGURE 23.

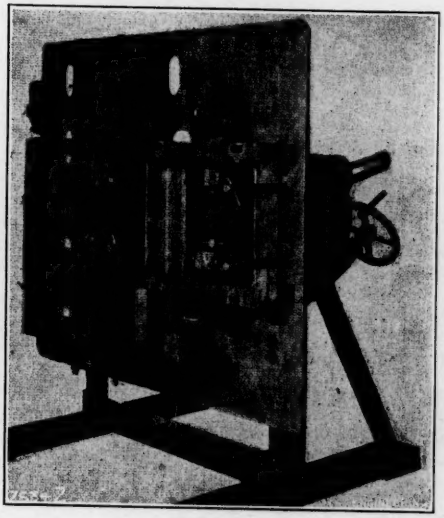


FIGURE 24.

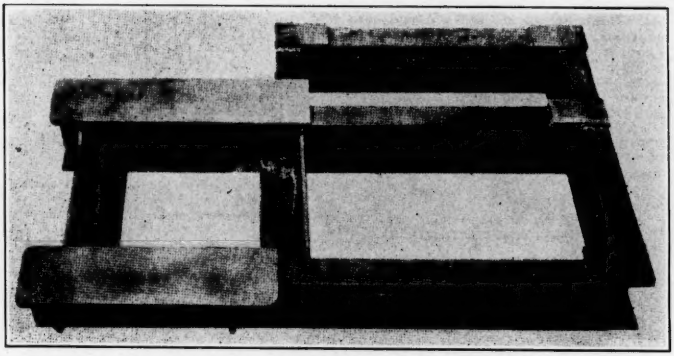
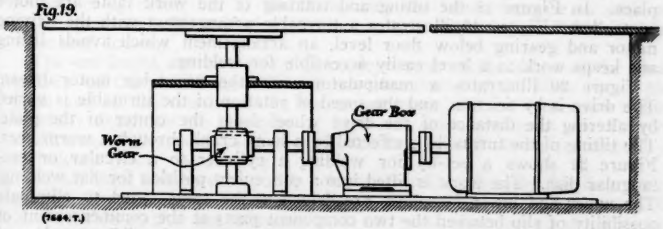


FIGURE 25.

where the two channels on which the rollers are mounted can be located at various points along the base, and the rollers themselves can be adjusted at different points along the cross members. This form of turning device is used for butt joints in all plate thicknesses or lap joints in fairly thin plate. A rotating device of this type is improved considerably by the addition of a variable-speed motor drive. For lap joints in thick plate it is preferable to tilt the work as shown in Figure 3, which shows motor-driven

Fig.19.



(7884.T.)

Fig.20.

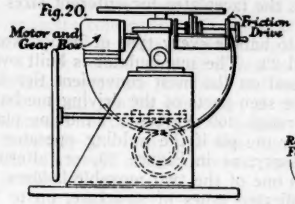
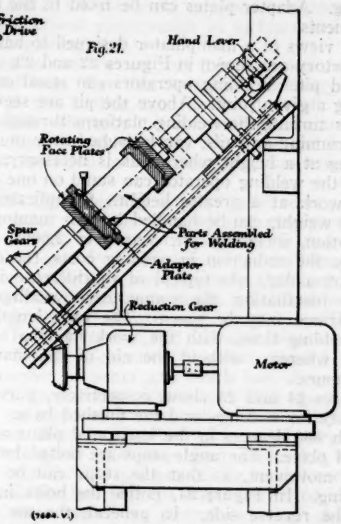


Fig.21.



(7884.V.)

rollers on a tilting table. Speed of rotation is determined by a variable-speed motor with remote control, connected by a flexible cable which can be placed near the operator. Figure 4 shows tilting-roller equipment for welding heavier work. While actually built for automatic welding, this equipment is also used for hand welding.

Manipulators are designed to reduce handling time to a minimum, eliminating the need for labor and cranes, and to save floor space and increase safety by avoiding the need for packing blocks. They should be capable

of turning the job at least from the horizontal to the vertical position, and should have adequate locking mechanism to hold the work table in any required position. Figure 16 shows an armature frame on a welding manipulator. Figure 17 shows a number of small manipulators suitable for use in welding a variety of different types of work, up to about 15 cwt. Notice the reduction gear for tilting and the handwheel extended on its shaft well clear of the manipulator table, so that free access can be obtained to the handwheel if the job overhangs the table. A brake holds the rotating table in place. In Figure 18 the tilting and rotating of the work table are motor controlled. Figure 19 illustrates a turntable arrangement with the driving motor and gearing below floor level, an arrangement which avoids lifting and keeps work at a level easily accessible for welding.

Figure 20 illustrates a manipulator with the turn-table motor-driven. The drive is by friction, and the speed of rotation of the turntable is varied by altering the distance of the drive wheel from the center of the plate. The tilting of the turntable is effected by a hand crank through a worm gear. Figure 21 shows a set-up for welding a cylinder to a circular or rectangular dish. The work is tilted into a convenient position for flat welding. The upper and lower faceplates are driven at the same speed to eliminate possibility of slip between the two component parts at the commencement of welding. Adapter plates can be fixed to the faceplates for different sizes of components.

Two views of a manipulator designed to handle every type of heavy work in a factory are shown in Figures 22 and 23. The manipulator is built over a tiered pit. Welding operators can stand on the most convenient tier for welding a given joint. Above the pit are seen parts of the driving mechanism for turning the welding platform through 360 degrees. A moving platform, running on rails, can be puhed over the pit if the welding operator is working at a height where this is necessary, as in Figure 23, or, alternatively, the welding operator can stand on one of the two movable ladders to reach work at a greater height. Complicated types of structure, up to 10 tons in weight, can be handled by this manipulator. The initial cost of the installation, including pit, was over 2,000l., which was recovered in one year by the reduction in welding costs, the equipment being kept working 24 hours a day. As typical of the increase in speed of production effected by the installation, the comparative welding times for fabricating a machine frame may be given. The total length of the welds was 575 feet. The welding time, with the work mounted on the manipulator, was 77.9 hours; whereas, without the aid of the manipulator, the time taken was 113.4 hours.

Figures 24 and 25 show, respectively, part of a base plate assembled in jig ready for welding, and the finished base. The table of the jig revolves through 360 degrees in the horizontal plane and through 240 degrees in the vertical plane. The angle stops are bolted into elongated holes to allow for lateral movement, so that the stops can be moved aside for loading and unloading. In Figure 24, notice the holes in the table, to permit welding from the reverse side. In general, the use of jigs and manipulators for welded work will speed up production and avoid skilled welders doing work which can be undertaken by unskilled labor.

END LOSSES OF TURBINE BLADES.

According to A. Meldahl, whose article in the November, 1941, Brown Boveri Review is reprinted below, clearance loss by no means represents the total end losses of turbine blades. Total end losses are said to depend

on shape of the blade end, profile and sharpening of the blade tip, pitch, type of flow and turbulence before the blading as well as clearance. The author suggests a new formula for the efficiency of turbine blading which takes into account the theory of similarity. The formula agrees well with measured values.

I. Introduction.

At some distance from the ends of the blades of a grid, only the pure profile losses occur, but at the blade ends the losses increase to such an extent that the efficiency of a turbine is to a high degree determined by these end losses.

The end losses are in the first place dependent on the shape of the blade end, *i.e.* on the profile together with any sharpening of the blade tip, on the pitch and on the clearance. In the second place, the end losses are fundamentally dependent on the type of flow and consequently on the Parsons coefficient, on Reynolds' and Mach's numbers as well as on the degree of turbulence before the blading.

Of all these influences, actually only that of the *clearance* has been more closely observed up to the present time. This is easy to understand when it is considered that the influence of an increased clearance on the efficiency makes itself felt very clearly and in an extremely unpleasant manner, so that engineers concerned with the practical construction of turbines had to study this question from the very beginning.

On the other hand, it is not so easy to see that this *clearance loss* by no means represents the *total end loss.* For this it is first necessary to know the actual profile efficiency of the blading, and it is not at all simple to measure the actual, pure profile efficiency on a completed turbine. Consequently the clearance was regarded as the only source of end losses and the efficiency, obtained by extrapolation to zero clearance, was taken as the profile efficiency.

In the literature dealing with this subject it is practically never mentioned that considerable losses also occur at blade ends without clearance, e.g. on shrouded blading, on nozzle blades and on bladed rings as found in the Ljungström construction. That even Prof. Stodola in his well-known book does not touch on this question at all, may serve to show that such thoughts did not even enter the heads of the leading men in the field of steam turbine construction, because Stodola's work "Dampf- und Gasturbinen" contains, even today, a veritable fund of interesting and valuable observations and suggestions. The English "Nozzle Committee", which carried out tests on nozzle segments over a number of years and which had adequate means at its disposal, also preserves an absolute silence over this question in its reports, a fact which is all the more surprising when it is considered that the "Nozzle Committee" was chiefly concerned with nozzle segments of small height, that is with blade grids with very short blades for which the end losses represent a quite considerable portion of the total losses.*

The modern theory of circulation has perhaps also contributed its part to veiling the end losses. In the case of an aeroplane wing, the "lifting vortex" round the wing must be cast off at the wing tip, and the trailing, marginal vortices which result, provide a very clear explanation of the tip losses. In the case of bladed grids with blades bounded at both ends, on the other hand, the circulation round individual blades, in accordance with the usual, simplified method of treatment, can close "by penetration through the end walls" external to the space where flow takes place, so that apparently no vortices at all can separate and cause end losses. The pure

* 50 per cent and more.

surface friction of the end walls is, moreover, not appreciable. This gives rise to the impression that, apart from the clearance losses, the end losses are negligible.

This opinion could be maintained so long as the actual, pure profile efficiency of the usual turbine blades was not exactly known. It has, however, already been found possible, a number of years ago, to measure the profile loss of an aerofoil, in a comparatively simple manner, by measurement of the pressure head in the wake behind the wing.* We have made similar measurements on models of turbine blades with considerable success.† In the course of these tests it soon became clear that the profile efficiencies determined in this manner were considerably higher than those obtained from turbine tests by extrapolation to zero clearance. The difference could only be due to additional end losses, and it was consequently necessary to examine this question more thoroughly.

II. Theory.

A great deal has been published on the tip losses of aeroplane wings with free ends, but the sources of information for aerofoils with a solid boundary at their ends are few and far between. Nevertheless, an investigation carried out by Betz ‡ yields really interesting information.

Betz gives the following formula for the *minimum* induced drag due to flow in a clearance gap at the tip of a wing

$$W = k_2 \cdot \frac{A^2}{\varrho \cdot w^2/2 \cdot l^2} \qquad (1)$$

W = induced drag

A = lift

l = blade length

w = mean, relative velocity of flow

σ = density

k_2 is a form factor which can be calculated from the following formulae:—

$$k_2 = \frac{K(x)}{8 \cdot K'(x)} \qquad (2)$$

where $K(x)$ and $K'(x)$ are the complete, elliptic integrals

$$K(x) = \int_0^{\frac{\pi}{2}} \frac{d\varphi}{\sqrt{1 - x^2 \sin^2 \varphi}} \qquad (3)$$

$$K'(x) = \int_0^{\frac{\pi}{2}} \frac{d\varphi}{\sqrt{1 - (1 - x^2) \sin^2 \varphi}}$$

* See, for example, Durand "Aerodynamic Theory."
† Brown Boveri Review, January/February 1938.
‡ "Hydraulische Probleme" p. 161 seq.

and where the variable x has the form

$$x = \tanh\left(\frac{\pi \cdot s}{t \cdot \sin \alpha_2}\right)$$

s = clearance
t = pitch
α_2 = relative exit angle

This formula of Betz can be converted in the following manner. Instead of k_2, a function σ is introduced which is defined by

$$\sigma \equiv \frac{\pi}{\log_e 2}\left(4\,k_2 - \frac{s}{t \cdot \sin \alpha_2}\right) \qquad (4)$$

The function σ disappears for $s = 0$, and increases very rapidly with increasing s, approaching the value 1 asymptotically (Figure 1). For all practical purposes we may here put $\sigma \cong 1$, under which circumstances

$$k_2 \cong \frac{1}{4}\left(\frac{s}{t \cdot \sin \alpha_2} + \frac{\log_e 2}{\pi}\right) \qquad (5)$$

If, now, formula (1) is applied to symmetrical turbine blading, we obtain by calculation the expression

$$\Delta \cong \frac{\Delta w_t \cdot \bar{w}}{u \cdot w_a}\left[\frac{s}{l \cdot \sin \alpha_2} + \frac{\log_e 2}{\pi} \cdot \frac{t}{l}\right] \qquad (6)$$

where

Δ = power loss referred to the output
Δw_t = alteration of the circumferential component of the velocity
$\bar{w}$ = mean velocity $= \left(\vec{w}_1 + \vec{w}_2\right)/2$
w_a = axial component of the velocity
u = circumferential velocity.

The first term in the brackets is obviously the clearance loss. The second term is an additional loss which, within practical limits, is independent of the clearance but which exists, at least theoretically, only because a clearance is actually present. This fixed end loss is proportional to the pitch.

Formula (6) can be rewritten in different forms. Two such conversions may be briefly described here, since they provide interesting fundamental conceptions.

1. Formula (6) may be multiplied and divided by $1/D$ (D = mean diameter of the blading). Since $\pi D/t = n$ = number of blades, we have in the first place

$$\Delta \cong \frac{\Delta w_t}{u} \cdot \frac{\bar{w}}{w_a} \cdot \frac{D}{l} \left(\frac{s}{D} \frac{1}{\sin \alpha_2} + \frac{\log_e 2}{n} \right) \quad (7)$$

Further $w_a \cdot l/D = V/\pi D^2 = c/4$, where V is the volume passed in unit time, and consequently c is that (fictitious) velocity with which the volume V would flow through an opening of circular cross-section with diameter D. Since, moreover, $\Delta w_t/u = gH/u^2$ and $u/2\bar{w} = \cos \bar{\alpha}$ ($\bar{\alpha}$ = mean change of direction of flow), we obtain the formula

$$\Delta \cong 2 \frac{gH}{uc} \cdot \frac{1}{\cos \bar{\alpha}} \left(\frac{s}{D} \cdot \frac{1}{\sin \alpha_2} + \frac{\log_e 2}{n} \right) \quad (8)$$

The blade length does not appear at all in this formula. Small end losses can, therefore, only be obtained by means of:—

(a) small heat drop per stage, H
(b) large circumferential velocity, u
(c) large fictitious velocity, c.

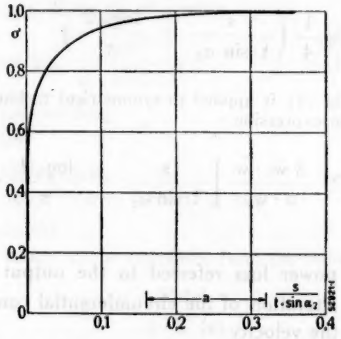


FIGURE 1.—BEHAVIOR OF THE FUNCTION.

$$\sigma = \frac{\pi}{\log_e 2} \cdot \left[4 k_2 - \frac{s}{t \cdot \sin \alpha_2} \right]$$

The figure shows it is sufficiently accurate to put $\sigma = 1$ in the range occurring in practice (which is indicated by $\leftarrow a \rightarrow$). It then follows that:

$$k_2 \cong \frac{1}{4} \left[\frac{s}{t \cdot \sin \alpha_2} + \frac{\log_e 2}{\pi} \right]$$

So far as the blade angle is concerned, the following statements apply. In the case of blades having small curvature—the calculation according to the Betz formula is only valid for such blades—$\bar{\alpha} \cong \alpha_2$. The first term in the brackets is, therefore, a minimum for $\bar{\alpha} \cong 45$ degrees. Since the profile losses are also a minimum for $\bar{\alpha} \cong 45$ degrees, the most favorable value will be somewhat less than 45 degrees due to the influence of the second term in the brackets.

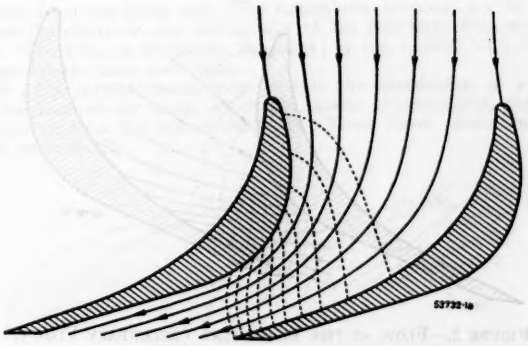


FIGURE 2.—FLOW IN THE CHANNEL BETWEEN TWO BLADES.

——— = Streamlines.

- - - - - = Isobars (lines of equal pressure).

The pressure gradient is perpendicular to the isobars. The component parallel to the streamline accelerates the flow, while the component perpendicular to the streamline diverts the flow.

2. Another very interesting form is obtained if the *stressing of the blades by the centrifugal force* is introduced.

The tensile stress produced in a cylindrical blade is

$$\sigma_z = 2\frac{\gamma}{g} \cdot u^2 \cdot l/D$$

(γ = specific weight of the blade material)

$$\frac{l}{D} = \frac{1}{2} \cdot \frac{\sigma_z}{\gamma} \cdot \frac{g}{u^2} \qquad (9)$$

σ_z/γ represents here a length, and is indeed that length which, when freely suspended, would produce the same tensile stress at its upper end due to its own weight as is produced in the blade by the centrifugal force. This length—which is related to the "rupture length"—will here be represented by L. Since $w_a/\bar{w} = \sin\bar{\alpha}$, we have

$$\Delta = 2\frac{H}{L} \cdot \frac{1}{\sin\bar{\alpha}}\left(\frac{s}{D} \cdot \frac{1}{\sin\alpha_2} + \frac{\log_e 2}{n}\right) \qquad (10)$$

Since $\bar{\alpha}$ cannot be much greater than 45 degrees on account of the profile efficiency, and s/D is also fairly constant, formula (10) shows that the end losses can only be kept small by adopting a *small heat drop* and a *high stress in the blades.*

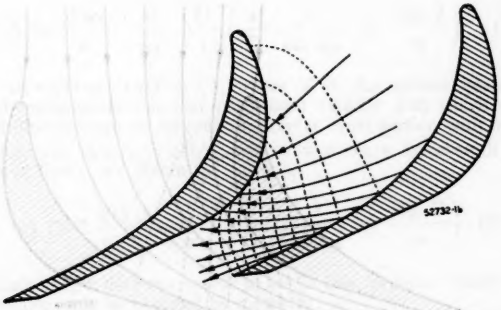


FIGURE 3.—FLOW AT THE BLADE END (SECONDARY FLOW).

——— = Streamlines.

- - - - - = Isobars (lines of equal pressure).

The isobars are the same as in Fig. 2. The streamlines follow the pressure gradient and are, therefore, everywhere perpendicular to the isobars.

The width of the blade appears in neither of the two formulae. Its only influence is that the number of blades n can be made larger for narrow blades without spoiling the profile efficiency.

If the clearance actually disappears completely, $\sigma = 0$ in equation (4) and the end losses likewise disappear in theory. In spite of this, tests show that considerable end losses, accompanied by the formation of vortices, still remain in existence. These end losses are due to secondary flow at the ends of the channel between the blades.

Figure 2 shows the undisturbed streamlines between two blades, and also, as dotted lines, the lines of equal pressure. The pressure drop is at right angles to the lines of equal pressure, the component in the direction of flow producing an acceleration, and the perpendicular component causing the flow to change direction.

At the ends of the blades the velocity is reduced to zero by the friction and there is, therefore, no centrifugal force. A flow is, therefore, produced in the immediate vicinity of the wall which is chiefly determined by the pressure drop alone, and the streamlines are thus perpendicular to the lines of equal pressure in this zone. Figure 3 shows these streamlines.

Figures 4 and 5 show that this secondary flow has a similar action to a clearance gap at the end of the blades. Figure 4 represents the free end of a blade. Since the pressure on the hollow side of the blade is greater, flow takes place round the free end of the blade. The streamlines on the hollow side spread out like a fan, while on the back of the blade they are forced away from the end. The streamlines coming from both sides of the blade are no longer parallel at the trailing edge, but cross one another. Vortices are, therefore, formed and cast off at the trailing edge, thus producing end losses.

Figure 5 shows the influence of the secondary flow at the end wall of the channel between two blades. For the sake of clearness, the end wall itself has been left out of the diagram. As may be seen, the diagram of the streamlines on the blade surface is quite similar; on the hollow side the streamlines spread out like a fan while on the back of the blade they are

forced away from the blade end. The streamlines, however, are no longer closed over the clearance gap, but by way of the secondary flow along the end wall. The effect is practically the same; at the trailing edge vortices are formed which cause end losses.

In both cases further losses occur due to the introduction of an alien flow on the back of the blade, which can, under circumstances, cause the flow to separate from the back of the blade. These losses cannot, however, be treated theoretically.

FIGURE 4.—FLOW AT THE FREE END OF A BLADE.

Flow takes place round the end of the blade because the pressure on the hollow side is greater than the pressure on the back of the blade. The streamlines spread out like a fan on the hollow side, and are forced together on the back. At the trailing edge the two flows cross one another and thus form trailing vortices.

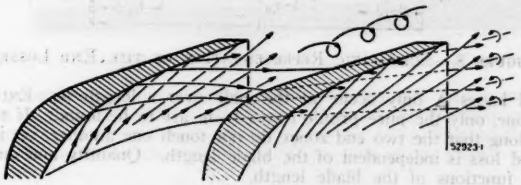


FIGURE 5.—SECONDARY FLOW AT THE END OF A CHANNEL BETWEEN TWO BLADES.

The end wall has been removed in order that the flow may be more clearly illustrated. The streamlines on the hollow side are drawn out to the form of a fan by the secondary flow, and on the back they are forced together. At the trailing edge the streamlines cross one another and form trailing vortices exactly as in the case of the free blade end.

III. TESTS.

The flow at the ends of blades can be easily observed on stationary models. In addition to the usual observations with Pitot tubes and streamers, the following method has proved to be very instructive. The blade is given a coat of a suitable oil paint, and is then subjected to the action of the air current while the paint is still wet. The paint is partly blown away and the streamlines in the immediate neighborhood of the walls are clearly reproduced in the wet paint on the blades and intermediate pieces. In this manner, the secondary flow at the blade ends, as deduced from the theory given above, can be rendered visible during the test (Figure 6).

In the accompanying photograph (Figure 7), the disturbed end zones can be distinctly seen in the coat of paint. When the pitch is small, only a very narrow interference zone is formed, which increases rapidly with increasing pitch. Although no conclusions can be drawn from these pictures as to the magnitude of the end losses, it is nevertheless quite obvious that the end losses must increase with increasing pitch, just as required by the theory.

It is not possible to determine the magnitude of the end losses with stationary models. The "induced losses" are by no means definitely lost immediately behind the wing; fundamentally speaking, they could be recovered.* The measurement of the energy immediately behind the blade model, *e.g.* with a Pitot tube, consequently yields only a very small end loss. The energy, however, is partially transformed into such a mixed state by the induced vortex, that it can no longer be usefully recovered in a normal row of blades; it is devaluated, although the degree of devaluation cannot be measured on a stationary model. Further tests were, therefore, carried out on an air-turbine and the results were evaluated on the following basis:—

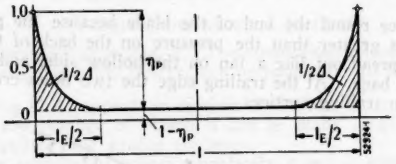


FIGURE 8.—SCHEMATIC REPRESENTATION OF THE END LOSSES.

The end losses Δ only occur at the ends over a length l_E. External to this zone, only the pure profile losses $1-\eta_p$ are to be found. If the blade is so long that the two end zones do not touch one another, i.e. if $l > l_E$, the end loss is independent of the blade length. Quantity and output are linear functions of the blade length.

So long as the blade is longer than the total interference length l_E at the ends (Figure 8), it can be assumed that the blade length has no influence on the end interference. If, therefore, two blade ends are geometrically similar as regards profile, pitch, sharpening and clearance, the interference at the end is a function of the geometrical dimensions enumerated, as well as of the type of flow, i.e. it is also a function of the Parsons coefficient and of the Reynolds and Mach numbers, but it is *independent of the blade length*. For otherwise identical conditions, both the quantity G passing through the turbine, and the output L are linear functions of the blade length, and the following expressions are valid:

$$G = g_b \cdot l/b + G_E \qquad (11)$$

$$L = G \cdot H \cdot \eta_p - L_E \qquad (12)$$

* When migratory birds, e.g. wild geese and cranes, fly in V-formation, the trailing birds make partial use of the vortex energy left behind in the air from the flapping of the wings of the leading birds. The foremost bird does not enjoy this advantage and is, therefore, relieved from time to time.

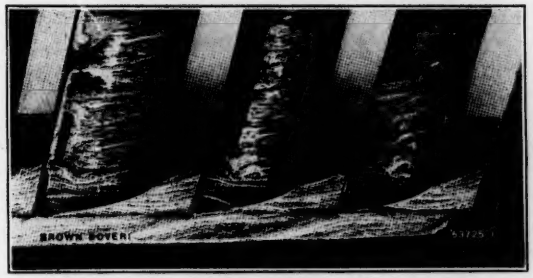


FIGURE 6.—SECONDARY FLOW AT THE END WALL OF A GRID OF BLADES.

The streamlines in the immediate vicinity of the wall are recorded on the coat of wet paint. On the back of the blades, the streamlines are forced together. It is noteworthy how exactly the streamlines at the end walls conform with the theoretically deduced streamlines shown in Fig. 3.

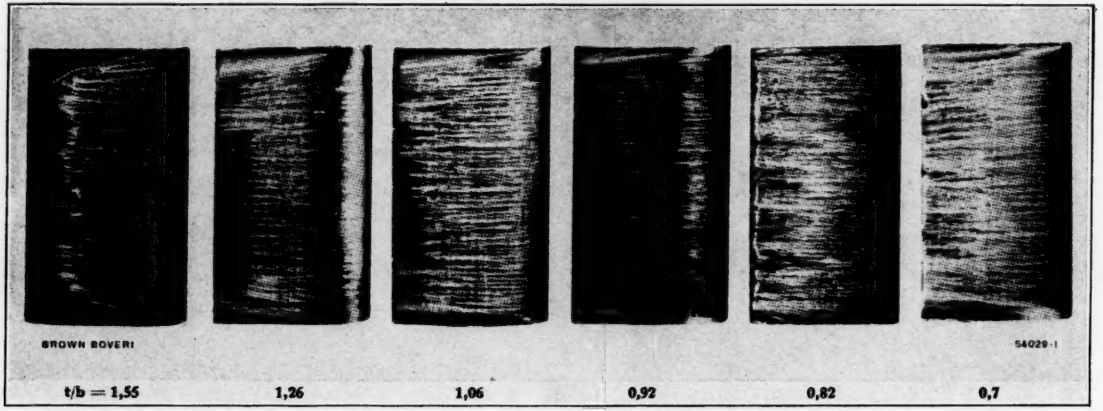


FIGURE 7.—INCREASE OF THE END LOSSES WITH INCREASING PITCH.

While the end interference is hardly visible for the smallest pitch, it extends far into the healthy flow in the case of the largest pitch.

l = blade length
b = blade width (serves as unit of length)
G = quantity flowing past the blade
G_E = additional quantity at the blade end
g_b = quantity per unit of length
L = output
L_E = end loss
H = heat drop
η_p = profile efficiency

With the abbreviations

$$\gamma_E = G_E/g_b \qquad (13)$$

$$\delta_E = L_E/g_b H \qquad (14)$$

we obtain the following expression for the efficiency:—

$$\eta = \eta_p - \frac{\delta_E}{l/b + \gamma_E} \qquad (15)$$

The quantities δ_E (end loss) and γ_E (end quantity) are functions of the shape of the blade end and of the type of flow, but are *independent of the blade length.*

If measurements are made on blades of different lengths, which are otherwise the same, δ_E and γ_E can be determined. The measured values of G are first plotted in function of l/b, in order to find γ_E (Figure 9). Then the measured values of the output are likewise plotted in function of l/b in order to obtain δ_E and η_p (Figure 10).

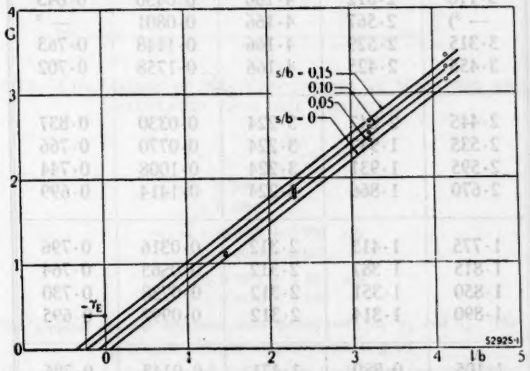


FIGURE 9.—DETERMINATION OF THE QUANTITY INCREASE AT THE BLADE END.

o = measured value (see table).

The lines represent the corrected values with the blade clearance s/b as parameter.

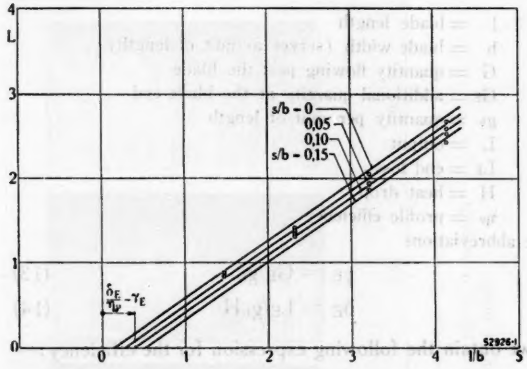


FIGURE 10.—DETERMINATION OF THE REDUCTION OF OUTPUT AT THE BLADE END.

o = measured value (see table).

The lines represent the corrected values with the blade clearance s/b as parameter.

Quantity [1] G	Output [1] L	Blade length [2] l/b	Clearance [2] s/b	Efficiency η
3·170	2·672	4·166	0·0430	0·843
— [3])	2·567	4·166	0·0801	— [3]
3·315	2·529	4·166	0·1148	0·763
3·455	2·425	4·166	0·1758	0·702
2·445	2·047	3·224	0·0330	0·837
2·535	1·942	3·224	0·0770	0·766
2·595	1·931	3·224	0·1008	0·744
2·670	1·866	3·224	0·1414	0·699
1·775	1·413	2·312	0·0316	0·796
1·815	1·387	2·312	0·0563	0·764
1·850	1·351	2·312	0·0738	0·730
1·890	1·314	2·312	0·0992	0·695
1·105	0·880	1·471	0·0143	0·796
1·112	0·838	1·471	0·0291	0·748

[1] Values for purposes of comparison, reduced to unit heat drop.
[2] b is the blade width and serves as the unit of length
[3] The measurement of quantity proved to be inexact.

On account of the unavoidable errors in the observed values, it is definitely well worth while to employ the method of least squares for correcting the readings, because this method gives not only the most probable values of the desired quantities but also the mean value of the error. The correction by the Gauss method thus provides protection against an unjustified faith in the exactness of the results obtained. The table contains a complete series of measurements on a *single stage* of reaction blading. The efficiency has been determined in function of the Parsons coefficient for four different blade lengths.

Figure 11 shows the results obtained by calculation from the observed data. Despite the fact that the simplest possible equation was assumed, the measured points lie in very close proximity to the calculated curves.

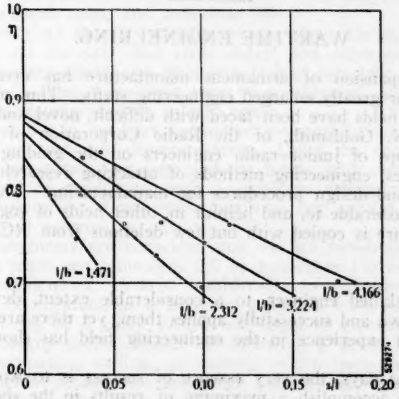


FIGURE 11.—COMPARISON OF THE EFFICIENCIES AS MEASURED AND AS CALCULATED FROM THE FORMULA.

$$\eta = \eta_p - \frac{\delta_E}{l/b + \gamma_E}$$

$$\eta_p = \begin{array}{l} 0{\cdot}9091 \\ \pm\, 0\ 0069 \end{array}$$

$$\gamma_E = \begin{array}{l} -\,0{\cdot}0422 + 2{\cdot}790 \quad s/b \\ \pm\, 0{\cdot}0037 \pm 0{\cdot}074 \end{array}$$

$$\delta_E = \begin{array}{l} +\,0{\cdot}1011 + 4\ 667 \quad s/b \\ \pm\, 0{\cdot}0118 \pm 0\ 158 \end{array}$$

The simplest possible equations were chosen both for γ_E and δ_E. The agreement with the measurements is good

(The numerical values are only valid for the single-stage reaction turbine tested!)

Especially noteworthy is the expression found for the end losses

$$\delta_E = 0{\cdot}1011 + 4{\cdot}667 \text{ s/b.}$$

Since s/b, in practice, varies between the limits 0.04 and 0.08, the pure clearance loss, i.e. the loss dependent on the clearance, actually represents only about three quarters of the total loss. There remains, therefore, a quite considerable *pure end loss* independent of the clearance, which appreciably reduces the efficiency of short blades even for the smallest clearance.

As previously mentioned, the end losses are generally first irretrievably lost in the next row of blades. Consequently it is not possible to apply the results of tests on single-stage turbines directly to multistage turbines, but tests must be carried out for different numbers of stages, from the results of which the losses of multistage turbines may then be obtained by extrapolation.

WARTIME ENGINEERING.

The rapid expansion of armament manufacture has created a corresponding need for greatly enlarged engineering staffs. Thus many engineers working in new fields have been faced with difficult, novel and urgent problems. Dr. A. N. Goldsmith, of the Radio Corporation of America, has lectured to groups of junior radio engineers on the guiding principles of engineering ethics, engineering methods of attacking research and development problems and design procedures for manufacturing. The included information is transferable to, and helpful in, other fields of engineering. Dr. Goldsmith's lecture is copied with but few deletions from RCA Review for April, 1942.

While every skilled engineer, to a considerable extent, develops certain methods of his own and successfully applies them, yet there are some general rules which long experience in the engineering field has shown to be safe guides.

In these trying days, the very essence of success is to work at highest efficiency and to accomplish a maximum of results in the shortest possible time. This is a test period for every engineer, just as it is for our country. Loyalty to our country, to the organization for which we work, to our fellow engineers, and to ourselves must alike prompt us to make the best possible use of every minute of time and to produce the maximum possible output of high quality.

In normal times a moderate degree of indulgence in certain personal limitations or weaknesses might occasionally be tolerated. But in these days of terrific emergency and extreme stress, grim necessity urgently compels the stern repression *by each man within himself* of carelessness, of laziness, of thoughtlessness, and of uncooperativeness.

Engineering Ethics.

As an engineer, each of us is a member of a *profession* and not a trade. That is, we are members of an intellectual fellowship rather than a competitive commercial group. It is important always to remember this in determining our attitude toward our work, our fellows, our company, and our country. As professional men, we have been highly trained over a long period of years. The knowledge of the past has been placed before us in clean-cut form. Carefully written text books, painstakingly prepared lectures, the facilities of a university, and all the experience of a great organization have been put at our disposal. Thus you and I owe a great deal to our engineering predecessors, to our families, and to ourselves. And we

have the heavy obligation of achieving thorough effectiveness in our work and maintaining unusually high personal standards of conduct and performance. While it would be well if all men were to live up to such standards, yet in our case particularly is it essential that we shall maintain high professional and personal standing.

Under the conditions which we now face, we will find that we have specifically the following loyalties:

First, we owe a great debt to the United States, our own country, which is now engaged in a life-and-death struggle with destructive and ruinous forces stemming backward from stark savagery and barbarism. These forces represent a viewpoint which is the antithesis of freedom of thought, integrity and dignity of the individual, and the granting of opportunity to each of us for self-advancement and personal growth. They would represent the stultification and destruction of all that we have been taught to accept as the basis of an ordered, calm, and ambitious life.

The emergency we face is indeed a serious one. Through a combination of incredulity on our part that such violent and unscrupulous aims could exist in our modern world, and through a lack of foresight, "it is much later than we think." Thus time must now be compressed, and extraordinarily much must be accomplished in a brief space.

But this does not in the least mean that we must hasten hysterically, fall into a panic, or rush into disorganization. Quite the contrary. It means only that we must face our jobs and the future with steadfast bravery, a cool acceptance of conditions, and a stubborn determination which will overcome all obstacles. Given these qualities, our success and the victory of our country is assured.

While we engineers are conscious that we are members of a profession rather than a trade, we sometimes fail to remember the corresponding distinctions and obligations. A brief consideration of definitions is worthwhile. A trade may be regarded as a business, a special form of handicraft, or a pursuit or activity of a fairly skilled but usual or customary type. There are generally many people capable of following a given trade. A profession is an occupation, calling, or activity which necessarily does require exceptional mental attainments and other qualities and involves special discipline, both during the training period and during later professional life. Relatively few persons are provided by nature with the mental and moral equipment to succeed in a profession.

When we examine the differences between a profession and a trade still more closely, we are impressed by the gap between the two—perhaps a wider gap than most of us have realized. The length of time required for training and the scope and demands of the training are both greater for a profession than for a trade. The training of the professional man is indeed long, arduous, and costly. For many years he must acquire knowledge of numerous facts, of methods and procedure, and in many instances must achieve manual dexterity. He must develop habits of thought characterized by exactitude, careful reasoning, and willingness to face the facts, no matter how discouraging or unflattering to himself personally this process may be. A combination of knowledge *and* wisdom is necessary. And these are very different assets, for knowledge, by itself, may be far removed from that skill in its selection, and that judgment as to its timeliness and suitability of application, which are a part of wisdom.

Accordingly the professional man must pass through many years of severe training and must develop a broad viewpoint as to his work and a willingness to serve humanity in the practice of his profession which are less requisite in other paths of life.

The mental qualifications of the competent professional man are numerous and severe. He must be thoughtful, analytic, thorough, honest in his appraisals, and considerably above the average in his mental capabilities.

The character requirements of the true professional man are also numerous. Pre-eminently, he must have high ethical concepts, for it has been found that men who think obliquely and evasively are poor professional material. The professional man is also distinguished by a sense of his commitments to society, to his professional colleagues, and to his own standing. He usually has a feeling of solidarity with his fellows—and this is more on mental, moral, and social bases than from a purely material standpoint. The professional man is self-disciplined. He has a sense of restraint which prevents him from acting precipitately or thoughtlessly or from rushing to hasty conclusions. He has the strength of character to view himself as nearly impersonally as possible and to appraise alike his weaknesses and his elements of strength. At his best, the professional man has a loftiness of attitude which ensures the progress of his field and of himself. He should be more controlled than most men by such ethical concepts as fairness, justice, cooperativeness, and honesty of purpose in his actions.

It may seem to some of us that these ideals are almost unrealizable and perhaps largely theoretical. Yet such is not the case. The viewpoints just expressed have been developed during the long experience of the professions through the centuries and have been found necessary in practice if those professions are to endure and their practitioners are to prosper in high repute.

Professional men are set somewhat aside from the remainder of humanity by their specialized skills, and are accordingly not too well understood. They are also attractive targets for criticism from demogogic politicians, muddled reasoners, and other parasitic or controversy-seeking elements in the community. Against attacks from such groups the professional man has only his own standing and the repute in which he is held in the community as a shield against injury or destruction. It is therefore particularly incumbent upon him to practice so far as he can the qualities which have just been outlined. These needs are not mere idle imaginings. These characteristics of the true professional man are rather the results of experience through the ages. They may even be regarded as proven empirically necessary. They have certainly been shown by pragmatic tests to be part of the outlook and code of successful professional men. The pragmatic test is merely finding the answer to the question: "Does it work?" We may be assured that the highest professional standards *do* work—and indeed, their absence is a fatal defect in the case of the would-be engineer. I have not dwelt on the inner satisfaction that the better type of man achieves in living a finer sort of life along the lines of his chosen profession. Yet as the years pass, each one of us finds an inner satisfaction and strength in having pursued, sometimes in the face of discouraging difficulties, a pathway which followed the code of his profession.

I would not have you believe that I am depreciating trades or those who work in them. To the extent that men in commerce or trades elevate their standards, they too can share in the fine standing, worthy accomplishments, and inner pleasure that come to the true professional man. But those standards, which are perhaps only permissive (though preferable) in the present stage of our civilization when applied to men in general, must be regarded as mandatory for us in the engineering field.

Toward our own profession, every one of us owes the duty of carrying on a high tradition. Remember that we have inherited the knowledge and experience of the past, accumulated by painful toil, courage in the face of obstacles, and hard and straight thinking over the centuries. It is stimulating to recall that men have endured disbelief, scorn, poverty, intolerance, and even death so that your heritage and mine of knowledge and the capability of mastery over the forces of nature might exist. The least we can now

do is to show our appreciation of that priceless heritage. We must forever realize that we are members of a fraternity of intellectual workers.

Toward our own company we owe much the same loyalty as each man on a college team owes to his fellows. Enlightened executives—and there are many of them—have a deep interest in the engineer and abiding faith in his capabilities. These industrial leaders plan the help that can be given to our country through engineering skill. They then promise the Government definite accomplishments based on their faith in what we can do. We owe it to the country and to these men to make good.

As I have said, we are members of a fraternity of professional men. Toward our colleagues we owe frankness, true cooperation, clarity of expression, and unselfish teamwork. The major difference between an army and a mob is in its spirit and direction. Given capable direction and the spirit of unity, an army may confidently look toward victory. Each of you is no less in the front line of defense and offense than the soldier or sailor. We all know well that it takes considerable strength of character and broadness of viewpoint to suppress our little personal weaknesses and petty reactions. Yet it must be done, and this is the very time to resolve that it shall be done. The great engineer is he who does this job of real cooperation the best.

The Engineering Virtues.

Every engineer from time to time should try to assess himself frankly and honestly. Self-analysis and self-judgment are stepping stones to greater strength and wider achievement. In wartime or peacetime, the esteem of our fellows and our success in a worldly sense will largely depend on the extent to which we possess and practice certain qualities and methods. There is no use in refusing to analyze ourselves. Our fellow workers and directors will do so in any case, and there is no purpose in trying the ostrich trick of hiding one's head in the sand. Better to determine one's limitations, vigorously try to remove them, and then to walk with our heads high.

The following are some of the major engineering virtues:

Initiative.—Life may almost be defined as self-willed motion. When motion stops, life dwindles. Unless a man is ready to "start something" he will get nowhere. Lethargy, uncertainty, indifference, delay, and fear are paralyzing. Enterprise and keen thinking and fast action are the keys to success. Don't be too conservative in trying things out. Remember that a conservative has been humorously defined, with an undertone of indictment, as "a man who doesn't believe anything should be tried the first time." The great rewards of history, as well as inner satisfaction, often spring from trying it the first time.

Application.—Steady work is an amazing instrument for achieving results. Sweat is the best possible lubricant to keep the wheels rotating. Mere ideas in the abstract lead hardly anywhere. To get results, it is necessary to keep going, and planning, and working even when one is very weary and there is a great temptation to sit back and "take it easy." This last is a fatal fault in an engineer. It is inconsistent with our dignity, our loyalties, and our future success.

There is no good reason for worrying too much about toil. Relatively few people have been ruined by hard work but many have failed through laziness. Lack of application is a costly national or personal luxury. Maybe we have had too much of it in the past; but certainly now is not the time for it.

Originality.—Doing the same thing over and over is well enough in its way, but it is not enough during times of stress when unusual results are

necessary. Then originality becomes particularly important. Practice imaginative thinking. If you have an idea, carefully cultivate it in detail. Then try to find flaws in it, viewing it with real detachment and in a critical mood. Try to think up numerous alternative ways of accomplishing the same thing. Then compare the various ideas which occur to you as to their respective and comparative merits and faults. Such comparisons lead to a wise and practical decision. Learn to think creatively and in a prolific way. Any man can expand his capabilities in these directions by trying, just as he can develop stronger muscles by exercise. Never be afraid to discard ideas which seem inappropriate or faulty, or to accept new ideas, even though radical, if they seem necessary and practical.

Frankness.—One of the worst faults that an engineer can have is vagueness, the concealment of facts, or the lack of courage to face facts. Avoid silence where the communication of information is required. And avoid loose or incomplete information where definite statements are needed. We should try to tell the whole story. Science and engineering need the "truth, the whole truth, and nothing but the truth." Substitute candor for double talk, which latter is alike the bane of engineering, politics, and many another field. Engineering does not need "verbal glamor boys"; it demands really creative workers with a genuine output.

Personal Relations.—As we all know only too well, it is easy to develop the fault of seeing no good in the ideas or work of the other man. We engineers should keep an open mind. Let us listen very calmly, coolly, and judicially to the other man's ideas. Think how you yourself would react to a scornful, unfriendly, or close-minded reaction to your own ideas. Try to find something valuable in the other man's proposals, for your own sake as well as his. If you must disagree, after careful consideration, do so courteously and clearly. Explain exactly why you disagree and how far you disagree, and give the other man a full opportunity to convince you with his arguments and reasoning. There is relatively little danger in being open-minded but much hazard in keeping our minds shut.

To get on in life, we must always remember that we are part of a community made up of many diverse elements. Think as well as possible of the other man and of his ideas. And, above all, avoid backbiting. It may be easy to injure the reputation and standing of the other fellow and (with regret and shame be it admitted) there is occasionally a human temptation to play tricks of this kind. But remember that it is probably equally easy for him to hurt your standing. It may be stressed that men generally do not admire the engineer who selfishly depreciates his fellow workers and their accomplishments. This is the easy way to lose friends, standing, and self-respect. Such attacks hurt his fellows, himself, and the general standing of his profession. Our careers depend in part on convincing the executives and others with whom we deal that engineers are broadminded, capable men who can do things, who are willing to do things, who think well of each other, and who can work together efficiently.

Relations With Your Chief.

You may safely take it for granted that those who are directing or supervising your work have been selected because they have had wide experience, have shown an unusual grasp and mastery of their subjects, and have proven that they know how to deal with emergencies and to appreciate good work. Give them your respect and loyal cooperation just as you will wish to receive such help and will need it in the future when you are in charge. Remember that, in the aggregate, they will be sternly judged by what you have accomplished; your company itself will be

appraised by the sum total of the efforts of its members; and our country will rise or fall on the integrated efforts and cooperation of all individuals and organizations in the United States.

Engineering Methods in General.

In order to reach a certain desired result, as engineers you may apply inductive methods, deductive methods, or both. Let us consider each of these and their most useful applications or scope.

Inductive Method.—This is the method where we reason from particular facts or observations to general rules or guiding principles. To carry out such a method it is first necessary to try various experiments. Thus, a number of specific or detailed things are done or methods tried or devices built. Then the results of the experiments or the performance of the devices are carefully studied (preferably with sufficiently exact measurement and test) to see if any underlying rule, law, or more general relationship can be detected.

There are many sorts of relationships from the simplest to the most complex. There are algebraic relationships, exponential or logarithmic relationships, trigonometric relationships, or even unclassified graphic relationships. Sometimes we will find that we must work by the "cut and try" method, determining by trial and error whether we have found a simple relationship between two or more quantities or elements in an experiment. It takes a good deal of skill and some experience to separate the essential from the unessential in such work and to discover whether some unforeseen or undesired factor is affecting the results and obscuring an otherwise simple or definite relationship. This is a case where we will find that there is no substitute for patience, thoughtfulness, and experience.

If it is thought finally that such a law or relationship has been found by the inductive method, it is checked again and again (though not necessarily immediately) by trying out experiments which develop from those which we have already completed, endeavoring to predict in advance the anticipated results by application of the supposed law; and then noting whether the facts and the theory still agree. If they do not agree, either the theory is wrong, or it is inapplicable to the particular case for some reason that must generally be determined speedily if further progress is to be made; or else the theory is incomplete. In the last-mentioned case, some important and contributing or controlling factor or factors have been omitted from consideration. These must be discovered if the law is to be expanded into a reliable guide. There is at least one other group of possibilities, namely that the experiment has been incorrectly performed or its results wrongly set down or its meaning misinterpreted.

In other words, if the relation which we have derived does not work out, we must not be content to say that theory and practice seem to disagree—we must try to find out *why* they disagree. A correct and complete theory cananot disagree with practice, although an incorrect or incomplete law will not be a useful guide. Often in finding out why theory and practice do not agree, you and I will learn valuable and helpful facts.

The inductive method is principally used by scientists and engineers in major or minor research projects where the experimenter is working at or near the boundaries of available knowledge and is trying to discover really new physical or chemical facts, methods, or laws.

Deductive Method—In applying this method we reason from the general law to the particular instance. Thus we start with a supposed

law, which has been found to be correctly applicable in the past and sufficiently complete and wide in scope to cover the assumed special instance under consideration. We then apply this law and predict a result or effect.

After the experiment is tried, we see whether the desired and predicted results are actually obtained. If so, we have another apparent strengthening of the law, and we have added confidence in its usefulness. If, however, the desired and predicted results are not obtained, we must make every attempt to find out why the law has seemingly failed. Merely abandoning the quest is generally insufficient. Perhaps the law was incorrectly applied. Maybe some other factors vitally affecting the results were omitted from consideration. Possibly the equipment under test was not correctly built or did not operate in the manner assumed by the designer. Or, finally, there may be a flaw or exception or limitation to the supposedly controlling law.

We should try to find out which is the case, for this will enable us to try the experiment the next time with a better chance of success or to determine how the law had best be modified or replaced in order to make it a safe and reliable guide.

Deductive methods, as might be anticipated, apply more usually in development work. Sometimes, when laws are well established as to their validity, scope, and completeness deductive work runs smoothly and to the satisfaction of the experimenter. However, this is usually the case only in long-established and well-covered fields. In general, even clever deductive reasoning requires caution and keen analysis at each stage.

Sometimes both inductive and deductive methods are used or mixed in handling the same problem. It generally takes an experienced worker with a wide knowledge of his subject, much experience, keen faculties of observation, and something of what we call (for want of a better term) "intuition" to handle such a problem by the use of the mixed methods. But it can be done and it frequently saves a great deal of time and effort. Intuition seems to be a sort of inner guidance or inspiration, dependent upon accumulated knowledge of a subject, and stimulated by the need for accomplishment and the urgent requirements of a situation.

It is a good idea in determining our methods to study carefully our own capabilities and preferences. As a general rule, the man who is most at home in research work and does it most naturally and best, is not so capable a development man as he likes to think he is. Similarly, the skilled development or design man who by instinct and experience develops equipment and methods readily may be a less effective pure research worker than he believes himself to be. For these reasons, every experimenter or designer should study his own capabilities carefully and impartially, if that is possible, and find out what type of work he does best. He should then endeavor to be assigned to that sort of work.

Further, if a man finds that he does his best work in a given sphere of activity, after he has carried any job to the point where it is about to pass out of that sphere into the next region, he should willingly turn it over to the next man for further work or completion. To speak in a blunt but friendly way, don't try to hog the development road—you may merely block traffic. Many a good research man has stuck to a job long after a development man should have taken it over and turned it into a commercially useful article. Often enough a development man has gone further into detailed design or manufacturing problems than is desirable or, on the other hand, has slipped back into research work where it would have been better to refer the unsolved problem again to a research specialist.

The advice I have just given does not mean that we should be without interest in any type of work other than our own. Quite the contrary, for it is an excellent idea to know something of the type of job which is done by the men who handle a problem before and after it reaches us. In that way we can best understand what is meant by the data or model or plan which reaches us and we can somewhat shape our own work to give our results or models to the next men in a form which will mean the most to them and will help them to carry on speedily. But avoid contracting that prevalent and contagious disease: "designer's itch." This grave ailment will make us satisfied with nothing that reaches us and will force us to try to change or re-design almost anything that passes before us. A little healthy appreciation of the other man's work and a broad attitude, applied daily to the mind, will be an effective remedy for this disease.

Engineering Methods of Attacking Problems.

Let us suppose that a problem has been submitted to us or a job assigned to us. What should be done next? While there are no general rules in the nature of a universal panacea that cures all ills, here are a few hints. In a long experience in such matters, they have proven to be sometimes useful.

In general, it is very helpful to start out by finding out *what is known.* That is, search for and thoroughly study the existing information on the subject. Don't depend too much on your memory for facts, figures, or methods. Even the most experienced engineers can easily enough forget very pertinent facts. I cannot too strongly stress the thought that it is no disgrace to have to seek information. Quite the contrary; for the search for information betokens the open and inquiring mind and the resolution to accomplish the desired results. As a matter of fact, it is sheer folly to insulate ourselves from sources of valuable data. Conversely, when anyone approaches us for information, let us try to help him patiently, remembering that we shall probably be in the same boat, and for the same good reason, before much time has passed.

There are a number of ways of finding out what is known which may prove useful to us. The simplest and most obvious one is first to go to the best available *text books.* We may get some general or specific data that way. The more nearly routine the problem, the more likely we are to find help in available texts. It is advisable to make notes at each stage of our study so that, when we have completed our investigation, our notes can be digested, summarized, and used as a guide. Great care and some patience is necessary in making notes. It does little harm to make notes too complete or elaborate, but it does a great deal of damage to have them so incomplete or brief that we have difficulty in interpreting them at a later date.

Let us see if the engineering *handbooks* have any available information on the devices or methods in which we are interested. If so, we shall make notes of them, entering a specific reference with each note so that we can re-locate the full information in the handbook or text book if we so desire. In fact, it is a good idea to give definite information as to where we located *any* data, in the form of a specific reference in our notes.

To get more reference material than is obtainable by the above methods, we should resort to the *published papers* in the major engineering journals. It is sometimes difficult to locate the desired references in an engineering journal. However, the annual or other indices of these journals will be of some help. The card index system of an engineering library may also

assist us. If we know the names of the engineers who have worked in a given field, we should look them up in the index or file so as to locate their papers which may bear on the subject at hand.

It is usually possible to accumulate a fairly full list of references on a given subject by a process of aggregation. One good reference article may give us a number of other references dealing with the same subject, which latter articles, in turn, will provide additional references. Thus we may soon accumulate a good *bibliography* of the subject and, in the process, a detailed knowledge of the particular field. The *reports* of our own organization's engineering staff are often helpful, and should be liberally consulted. They will usually contain more detailed and practical information than can be found in the average publication. *Standards reports* of the manufacturing associations (for example, the Radio Manufacturers Association or the National Electrical Manufacturers Association) or of engineering bodies (for example, the Institute of Radio Engineers, the American Institute of Electrical Engineers and the American Standards Association) may prove helpful. Now and then issued *patents or* accessible patent *applications* will also be instructive. But we should keep in mind that patents and patent applications are not necessarily completely scientific presentations in some cases nor will they always furnish the most useful forms of technical data. They should therefore be taken as interesting and suggestive but not necessarily final in every instance.

Speaking of reports, the technical reports prepared by you should be carefully written and in such detail as will enable others to understand fully what you have found and described. Technical report writing is a real art, and it is well to read and assimilate good books on the subject for your guidance.

It has been found that home study in the evenings, or even during part of the day, is sometimes not a bad idea at this stage of a new project. Frequently the change in surroundings or the quiet of the home may prove helpful in accomplishing a good deal in the assembly of data and in getting new ideas. It is usually a helpful plan to have loose-leaf note books for the data we shall accumulate or even to use a card index for particular subjects or fields in which we steadily work. These are useful tools for the engineer.

Continuing our process of finding out what is known, we must not hesitate to call on our *fellow engineers* (but try to avoid interrupting them at a particularly busy moment). There is no odium or loss of standing involved in asking questions. Nobody knows everything about a subject. Then too, we may be able to help the other fellow a little later on and he should be willing to give us what assistance he can at each stage.

In trying to get a definitely new idea, "hunch" or inspiration, don't force the issue too hard. Over-straining in an attempt to get a new idea may have the opposite effect. One can become stale or over-tired in that way. There are no rules governing the arrival of an inspiration. Some men get their best ideas in the early morning working hours, others in the late afternoon, and still others say they tend to get their new ideas during their normal hours of sleep, perhaps waking up with the answer to the problem which has been bothering them. It is not a bad idea to study your own reactions and to see into which of these classes you may fall. You can then try to arrange some of your work so that you can sit back and think intensely and take notes on our ideas at those times when experience shows you are most likely to "find the answer." It is thoroughly in order to make a great number of sketches and to jot down a considerable group of alternative ideas on any subject without analyzing each of these too thoroughly. Thus you will have a number of alternative plans in your notes, secured when you were in an "original

mood." You can always analyze each of these later for its advantages and defects thus choosing at last the one that seems most hopeful and likely to work out in practice.

Occasionally it is a good plan (if conditions permit) to lay aside work on a particular problem for awhile, and instead work for a spell on some other job. A return to the original problem in due course will sometimes show that considerable progress toward its solution has somehow been made in the intervening period.

And now, suppose that we have found what looks like a plausible or probable solution, method, or design. We shall then be ready to start putting our ideas into equipment, such as an experimental, functional, or "bread-board" model, or even what we hope will be the final manufacturing form. If we wish to make only a functional and experimental model we need consider only the requirement that it shall produce the desired result regardless of its adaptability to manufacture. But, if we are tackling the even more difficult job of making a manufacturing design—and this, it must be stressed, is an entirely different matter—we shall have to consider many additional problems of design as well as factory methods and limitations and economic controls. But in either case, this is the time to plan the model, of whatever type, very carefully. Consider in detail just what the model must accomplish. How can it be made most simply, and from the least possible number of parts? What parts are already available? Be sure to utilize these as a matter of economy of time and money. Even in building a purely experimental model, let us try to imagine as far as we can what the final and manufactured form will ultimately be like and then make our model as much like that as possible (unless we are in so early a stage of a development that it is not practical to visualize the final commercial form of the device). Keep our set-up as simple, complete, and reliable as possible; but let us not waste time on unimportant details. Cultivated good judgment along these lines is an important asset to the development and design engineers.

In making tests and observations on our set-up, we must watch not only for the desired results but also for any odd, or unexpected or undesirable effects or performances. Very close observation is important at this stage. A great many new things can be learned in that way and many future "headaches," or "bugs" in the apparatus, can be avoided. Your general slogan should be "watch closely and think hard." Avoid distractions and try not to hear or see what is going on around you unless it directly concerns your work. Develop, if you can, "earlids"—they are as useful in shutting out undesired sounds as are eyelids in keeping out intrusive scenes. Stop, look, and think!

There is another thought which may be particularly important to us in wartime development and design. It may sometimes seem to the engineer that the requirements of the Army, Navy, or Air Force are unnecessarily stringent or detailed. But it is well to remember that, while the customer may not be invariably right, it is an excellent idea to give him every benefit of the doubt. This is particularly the case in connection with military divisions of the government in wartime. They are closest to the actual use of the equipment under the stringent and grueling field conditions. They have had experience in the difficulties which arise in its use; they must live with it; and they generally will have a more complete and clear conception of service conditions which must be met than designers who have not been active in the field for a long time. Accordingly it is wise to keep an open mind on specifications even if they look too stiff. However, it is in order to ask questions as to the reasons for them. And if we have unusual difficulty in meeting them,

perhaps the specifying group may be able to suggest a solution or even to provide for us a model of something fairly similar to what we are supposed to produce and thus help us to "make the grade." Don't hesitate to ask for help in such directions, particularly under the present emergency conditions. It is readily possible to be too "dignified" in such matters. You will often find that the customer can be a helpful friend.

Engineering Factors Guiding Design.

All products logically start with a clear concept of their purpose, general construction, and mode of use. It is a difficult task then to translate them from the mental field into the world of physical things. This process involves the element of skilled design.

Design itself may be regarded as a means to a certain end. Its procedure is controlled by the function or use to which the product is to be put, the availability and suitability of the materials which go into the product, and the methods of construction and test which are at hand.

The use to which a product is to be put necessarily determines its functional design. That is, unless the design from the very beginning is such that the device will produce the desired practical result, no progress is possible.

Nevertheless, a merely functional design, however interesting and encouraging, is usually not in itself of major practical importance. It must be possible to manufacture the desired article readily, at a reasonable cost, and within acceptable limits of time. The availability of suitable materials, and of methods of handling such materials, determine whether the proposed design is manufacturable and economic. There may be many methods of translating a functional design into a manufacturing design, but some of these will be too complicated, too costly, or too slow to meet the requirements.

There is no substitute for good judgment and careful analysis. Accordingly designers should avoid plunging hastily in a direction which, for the moment, seems attractive. They should rather carefully analyze a number of possible methods of passing from a functional design to a manufacturing design. They should then select that particular method which best fits the current conditions.

One of the worst mistakes an engineer can make is to start design work without carefully considering the numerous guiding factors governing the nature and use of the desired product and then translating these factors, perhaps in tabular fashion, into corresponding physical embodiments. This should be followed by a thoughtful comparison of the various possibilities and the selection of that one which, on the whole, offers the maximum advantages and the minimum disadvantages. Rarely, if ever, will any decision be free from disadvantages—but good judgment and the relative appraisal of advantages and disadvantages will go far toward contributing to the success of the engineer. It is natural enough to be impatient with delay and to desire to get started on a job immediately. But yielding to these natural desires is frequently costly in time and money, for the later rectification of early mistakes is a sorry and trying job. "A thought in time saves nine."

There are a great number of factors which markedly control or influence the design of a given piece of equipment. Some of the most important factors are the following.

Nature of Specifications. — Specifications are a sort of bible to the designing and planning engineer. Many an engineer has invited—and encountered—serious trouble by not reading the specifications sufficiently closely and then checking on reliable methods of meeting each

and every feature of the "specs." Take the specifications very seriously. If you don't expect to be able to meet them, it is better to say so in advance and try to have them modified or else secure such help as will enable you to meet them.

It is important to be sure that every element in the specs can be met before the job is started. Once the design is well advanced, it often becomes an appalling task to change even one element without scrambling many of the others. Delays and changes in the later stages of the work are extremely costly and time-consuming. And it must be remembered that now, more than in any other period in our history, time is of the essence. Peacetime standards may not be adequate for wartime needs. To repeat, let us take the specifications seriously, plan meeting them in advance, be sure that each element can be met before we start, if that is at all possible, and make every effort to avoid changes in the latter part of the development or design.

Conditions of Use.—Obviously it makes a great difference under what conditions equipment is to be used since these conditions will vitally affect its design and required performance. Such questions as the following arise. Will the device be subjected to marked temperature changes? And with what effects on the materials and operation of the device? Or to wide changes in barometric pressure (with possible consequent high-voltage insulation problems)? Or to high humidity? Or to heavy jars and violent vibrations? Or to sudden accelerations? Will it be used continuously or only intermittently? Must it stand up under frequent and even prolonged overload? Must it be handled in the dark, and how? How noisy are the likely locations of its use? Must the operating personnel wear heavy gloves? Is it fixed, portable, or transportable equipment? Is it required to work in any position? Will many assemblies of the device be made, or only a limited number? Must it fit into a given space, or pass through openings of a limited size? From which side or sides must it be operated? How shall it be installed or removed? How is it associated with adjacent or nearby equipment? It it a well-standardized type or is it likely to be changed shortly? Are weight considerations of prime importance? How skilled are the personnel who are likely to use the equipment? What service problems must be met in advance?

We should try to accumulate as complete a list as we can of such working conditions, and then to give them major consideration in design and construction, as well as subsequent test. These requirements are of interest even in experimental models but are of course vital in the manufacturing model and design.

Servicing.—Every piece of equipment may develop some trouble sooner or later, and often enough at an entirely inopportune or dangerous time. Hence it is generally necessary that servicing problems shall be reduced to a minimum. Breakdown might occur at a tragically wrong time and place. Accordingly parts should be as far as possible conveniently accessible, easily removed, and readily replaceable (if there is any likelihood of their failure). Appearance is generally less important in wartime equipment than great convenience of use and reliability. The training and methods of the personnel who will probably do this servicing must be considered in each instance. Let us not hesitate to ask questions in these regards if we do not have the facts. A question is less embarrassing than an apparatus breakdown.

Cost.—It is obviously necessary that the designer of equipment which is to be manufactured from his plans must be thoroughly acquainted not only with the design processes but their capabilities, limitations,

and costs. It takes years of experience to become a master in this field but everyone should make a good start. As apparently simple a factor as the permissible tolerance set by the designer for a given dimension, or the reference point or plane of such a tolerance, may make a device either practical or impractical from the viewpoint of the factory expert. Further, it may make the product economic, or, on the other hand, unsalable. The same comment holds for specification of materials, finishes, and fittings. Thus there must be very close cooperation between the designer and the factory men if there is to be speedy, uninterrupted, and economic production.

Drawings.—Most large plants have specifications for standard drafting practice. Such practice may differ from plant to plant, and it is necessary to be thoroughly acquainted with the practice in our own company and systematically to adhere to it. Not only the designer but also those engaged in development work should become thoroughly acquainted with this material since it dictates a line of thought and procedure which is useful to all and which may short-circuit otherwise costly and unnecessary misunderstandings or errors. It is extremely wasteful and dangerous to make "off-standard" drawings.

Interchangeability and Economy.—Most modern manufacturing, particularly on a mass-production basis, depends on some form of interchangeability. Interchangeability, in its most complete form, enables the assembly of a complete device from component parts selected at random from lots of each of such component parts. That is, an operative device will result from the assembly of any available components. The advantages of such a possibility, both at the plant and under servicing and replacement conditions, are obvious.

In some forms of interchangeability only the sub-assemblies can be assembled at random to form an operative whole. These sub-assemblies form normal groups or combinations of parts. Sometimes the same sub-assemblies can be used as parts of a whole group of devices, which again increases economy in manufacture.

Selective interchangeability is another type in which all component parts are classified or graded into groups. A selected group of one component will then combine with a correlated selected group of another component, and so on. The advantage of selective interchangeability is mainly the broader tolerance in manufacturing each component which then becomes acceptable. On the other hand, the components must be classified or graded, and complete interchangeability in servicing may no longer be possible (that is, a certain amount of modification or machine work may be required from the service men).

Economy and speed in manufacture are largely associated with a wise and corresponding degree of interchangeability. Very careful design of the manufacturing model is involved in interchangeability. As previously indicated, the manufacturing model will generally differ markedly from a merely experimental or functional model. The functional model need only carry out a purpose or requirement and perform a certain job. However, it may be in a form inappropriate for manufacture or normal use in practice. Needless to say, manufacturing models should be designed with the help of the factory men. The construction of such models, their assembly, and their performance should be checked systematically in the factory throughout the manufacturing process. Any errors in the manufacturing design or in the process specifications can thus be corrected, the records made complete and instructive, and the lessons thus learned can be embodied as improvements, or in the avoidance of errors in later designs.

In the modern plant, fixtures, specialized machine tools, and cleverly contrived gauges, all play their part in economical production and in the obtaining of that precision which is required for the desired type of interchangeability.

Materials, Processes, Finishes.—In his selection of these factors, the development man and the designer can prove himself to be either a skilled and valuable man or the opposite. The designer should ask himself many and searching questions as to the available choices in each case, the advantages and disadvantages of each of them from the manufacturing and customer viewpoint, the relation between each of them and the corresponding saving of time in production (with particular reference to the needs of war production), and the cost and labor requirements of each. Minimizing the demand for highly skilled and experienced labor is important at this time.

Unusual materials, processes, or finishes are strenuously to be avoided because of the delays, chances of error, and high costs they introduce. Only where they can be proven to be absolutely indispensable should they be considered.

There are usually available in any large factory assembled data on standard shop practice. Here again we have material which is of great value to the practical development or design engineer and which should be thoroughly understood and applied by him.

Overall Reliability.—One of the characteristics of most adequate designs is that they are "balanced." That is, every part of the device is designed from a given viewpoint as to cost, performance, and reliability —and the criteria apply alike to each part of the design (except perhaps a few parts which, for some special reason, require unusual precision or particular care or better materials than the rest). Designing for an overall equalized reliability permits the user to judge more accurately what will be the probable life of the device in actual use. It contributes to economy in manufacture. Further, it prevents some one weakness from unfavorably controlling or limiting the useful life of the device. It requires considerable experience and good judgment, however, to design equipment so that it will run something like the "one-horse shay."

In closing, I would like to suggest that we all go through a mental over-haul, so to speak, every few months. It is astonishingly easy to get into a rut, or to drift behind the times. Let us review the thoughts which have been laid before us, and the experience we have gained. We shall decide whether we have used both of these to the full. We should keep notes of our own on the experiences we have had and draw conclusions from them which will serve as further guiding ideas for our own use in the future.

And we should always remember that our work is being increasingly appreciated and judged to be of paramount national importance. In its leading editorial on December 26, 1941 "The New York Times" said:

"As this war develops we can be increasingly thankful that we need no 'propaganda' to convince ourselves of the necessity and justice of what we are doing. All argument was silenced at Pearl Harbor. More and more this nation will value the gifts of certainty and of unity which our enemies bestowed upon us by their very act of aggression. Few causes and few wars are as simple as this one has become. The defeat of the Axis Powers has become as well-defined and essential an objective as was, a generation ago, the completion of the Panama Canal.

"We suspect that it arouses a not wholly dissimilar emotion. We have grown up since 1898, when we looked for 'glory' in our brief and one-sided war with Spain; and even since 1917, when an exalted and

Utopian mood seemed necessary. *Today's war requires the industrial and engineering type of heroism and fortitude. It requires planning, patience and exactness.*

"We know that war itself is not glorious, though there is glory in some of the qualities which it reveals and utilizes. We shall give praise to every American, soldier, sailor or civilian, who forgets risk and hardship and does his duty, or more than his duty. But principally the mood of this nation at this moment, if we may judge by what is said, written and done, is reflected in a desire *to get on with the job, and get done with it.* One may well believe that this is a frame of mind more dangerous to our enemies than the fanaticism of Berlin, Rome or Tokyo is dangerous to us. For, after all, the fanatical phase passes and is always limited to a minority in any nation, whereas the sober determination to finish what has been begun will engage a majority to the end.

"We think there prevails in this country a grim resentment at the necessity of devoting wealth, ingenuity, labor, courage and precious human lives to the ugly task of sanitation which the Axis has, by its existence, created. We have been interrupted at our work. We want no more such interruptions, and, God helping us, we will have no more."

Fellow engineers, we are in the first line of defense and offense. Let us then *think straight, plan well and work hard* to bring for all of us a brighter tomorrow.

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

AIR RAID PRECAUTION MEASURES.—On December 9, 1941, the Institution of Mechanical Engineers, the Institution of Civil Engineers and the Institution of Electrical Engineers held a joint meeting to discuss air raid precaution measures as related to the engineering industry. Addresses were delivered by Professor J. D. Bernal, Professor J. F. Baker, and Major J. W. Martin, all technical officers of the British Ministry of Home Security. Summaries of their addresses are reprinted from the April, 1942, Journal and Proceedings, Institution of Mechanical Engineers, in the order in which their names appear above.

Effect of High Explosives on Structures.

An understanding of the mechanism of damage by high explosives is invaluable when designing means for minimizing such damage.

The subject has been studied in three ways: (1) laboratory work; (2) field work on full-scale experiments unsuited to the laboratory; (3) recording at site the effects of air raids and the effects of particular bombs on structures.

Observations of air raids reviewed in the light of large- and small-scale experiments have made it possible to predict with confidence from laboratory tests what will happen in the field. What may be called a practical science of explosives has thus been built up, and it is now possible to assess the strength of a structure required to resist a given weight of explosive under given conditions.

It is not possible here to give a quantitative picture, but a rough idea is possible of the character of damage (excluding fire).

The main effects of explosives are covered by describing the arrival of the bomb, and then the three chief mechanisms by which mines and bombs cause damage: (1) blast; (2) bomb fragments; and (3) earth shock when a bomb explodes in the ground.

Mines usually have a reduced speed of falling and depth of penetration; they are intended to explode outside buildings or on ground, and achieve their object entirely by blast.

Penetrating bombs range from the 50 Kg. type, used in large quantities, to large bombs exceeding 1000 Kg. All these are provided with cases sturdy enough to penetrate buildings.

A point to stress is not how far a bomb can penetrate a building or the ground, but how far it goes before exploding. This is determined not by the penetrating power, but essentially by the kind of fuse it carries.

Consider the explosion of the bomb itself. If this occurs in the air the expanding gases cause the case to swell, and so thin it to breaking point. The fragments of the case are not distributed evenly, but fly into three main groups: (1) those shot off at right-angles to the side of the bomb, forming the main fragment zone; (2) tail fragments, usually light; and (3) nose fragments, usually heavy.

A wall near an exploding bomb receives fragments from the main zone, and a hole may be punched through. Momentum of fragments is large, and massed fragments have been known to shift concrete blocks.

The released gases expand initially at a speed of about 6000 feet per second; their impact on the surrounding air produces a blast wave in which pressure rises instantaneously to its maximum value, falls off gradually, and is followed by a suction phase of longer duration but smaller magnitude. With increasing distance both pressure and suction fall off (the pressure more rapidly), and the durations increase. Blast pressures on structures are different from the pressures ordinarily met in engineering because they act for so short a time. Thus a blast peak pressure of 20 pounds per square inch has quite a different effect from a static pressure of 20 pounds per square inch.

It is possible to analyze the reaction of a structure to the sudden short pressure and to estimate fairly accurately how a wall of given construction will react to blast of any character, by means of a calculation using the equivalent static pressure. High-frequency systems become subject to a greater equivalent static pressure than those with low frequencies because those with high frequencies move more rapidly, thus absorbing more energy of the blast. A very heavy structure reflects practically all the blast wave, and the structure is little affected.

Damage to light structures can occur by a resonance effect if the half-period of the structure is of the same order as the time interval between maximum pressure and maximum suction, the elastic rebound of the structure coinciding with the suction phase of the blast.

The type of failure of buildings under blast depends on the distance from explosion. Houses near the bomb collapse, the walls being pushed away by the pressure phase. Farther away damage is caused by the suction phase. When the external pressure is reduced, the air confined within the building presses on the walls, and if the suction is strong enough wall may even be pushed out. The shielding effect of other buildings is marked, especially for the pressure phase, so that damage is usually confined to roofs and upper floors of more distant buildings.

Blast is most important when the explosion is confined, its multiple reflections from walls, floor, and roof prolong the period during which pressure acts. Since the amount of damage to the structure is determined by the impulse, *i.e.* the pressure multiplied by the time for which it acts, the increased duration due to reflection increases the destruction. This destruction is therefore diminished by the flying off of light roofs and, to a lesser degree, by the bursting of doors and windows.

Primary damage is due to the direct effect of blast pushing over part of the structure; secondary damage is caused by subsequent collapse of

other parts. With existing buildings not much can be done to diminish the primary damage, but a considerable reduction of secondary collapse can be obtained by proper strengthening. This can clearly be seen by comparing fully framed structures with those having load-bearing walls which often exhibit spreading or secondary collapse.

When a bomb explodes in the earth, the case breaks into a few large fragments and the expanding gases make a spherical cavity in the surrounding earth. At the same time a shock wave travels out in all directions. On arrival at the surface the wave is reflected as a tension wave, a conical scab being thrown up, leaving a crater. The type of crater depends on the depth of explosion. Debris may fall back leaving a visible crater much smaller than the true one, or the bomb may be so deep as merely to heave and crack the ground, the explosive gases escaping and leaving a "camouflet."

The shock wave in the ground travels at different velocities in different layers and at different depths, and at some distance from the explosion the originally simple shock wave becomes a complex train of waves. Near the bomb considerable displacements of surrounding earth occur. These are partly elastic and partly plastic, so that ground may be moved several inches while the final displacement shown is small. A point fairly near a bomb has a violent upward jerk followed by a comparatively slow horizontal movement outwards and a subsequent partial recovery. Both vertical and horizontal movements fall off rapidly with distance.

Near the bomb, buildings are damaged by plastic displacement, while farther away they are subjected to shaking, and collapse may occur through resonance effect, although damage is usually limited to cracking. The shock wave is transmitted into the buildings through the foundations and is reflected as a tension wave from any free surface. Thus roofs are often lifted and may be thrown clear of the walls in small buildings, and the wall farthest away from the explosion may collapse, although the nearest part of the building may be distorted but not seriously damaged.

The effects of ground shock are more serious for underground structures than for surface ones. In an underground structure, the force applied to the nearest wall is transmitted through the roof and floor, and caving-in may occur leading also to roof collapse. However, structures in which floor, walls, and roof are well tied together can undergo considerable distortion without complete collapse.

Design of Protective Structures and the Defense of Industry.

Principles of structural design for protecting industry from air attack have now emerged which, though unfamiliar in normal engineering practice, can claim recognition as rational. A bomb exploding near a structure liberates energy so violently, and for such a short time, that it is the dynamical properties of the structure that are tested, and strength under simple static loading becomes relatively unimportant. For effective resistance the resilience of the structure is what matters, coupled with provision against failure by shear. Moreover, while ductility in material and continuity in structural form are essentials, these qualities must be combined with mass if the structure takes a form, such as a wall, which will directly oppose the blast. This is because the maximum work per unit area of surface which can be done by blast on the lamina is inversely proportional to the weight of the lamina per unit area.

The means for pursuing the study of principles and their application have included testing up to full-size trials, observation, critical and statistical analysis of results of air attack on structures, and mathematical work on the physics of explosion. As the practical outcome of this re-

search it remains to show how protective structures may combine with economy of labor and of such materials as are available, the qualities of weight, resilience, and resistance to shear, and how industrial buildings may be similarly endowed.

Well-placed shelters for the protection of workers during the "alert" period are an important feature in industrial defense. A satisfactory form has been evolved identical in outward appearance with the original brick shelter but incorporating vertical steel reinforcing rods. The rods are carried into the roof to tie it down. Similarly the rods are best taken into the floor, producing in effect a box section. For ease of placing the reinforcement the "Quetta bond" of brickwork has been useful. Such shelters have proved highly satisfactory not only against bombs exploding on the ground surface but also in resistance to the earth movements set up by bombs exploding after penetrating some distance into the ground near the shelter.*

Structural design for strengthening floors to carry debris loading and for stability under earth shock has also been rationalized. Deflexions which would normally be regarded as excessive are permissible if the weight of the collapsing superstructure is sustained. The design must be redundant, continuity being provided wherever possible, and a ductile material used so that large deformations can occur without failure. In addition to supporting the estimated vertical debris load, a horizontal load of 200 pounds per foot run applied at ceiling level and acting on any side must be sustained. The total maximum stresses may exceed the usual working stresses by not more than 33⅓ per cent in cases where the excess is due solely to the horizontal loading.

In small dwellinghouses with wood joist floors a braced frame is recommended, built independently of the existing structure.†

The fully framed building, being highly resistant to collapse, is eminently suited for providing air raid shelter. On certain floors it affords a substantial degree of overhead protection, and panel walls of the usual thickness offer useful resistance to glancing blows. Fire-resisting framed construction gives useful security from the primary effects of a direct hit if adequate cover extends over the whole floor on which the protected accommodation is required. Any additional lateral walling required for the protected accommodation is unlikely to increase the stresses beyond the limitations of British Standard Specification No. 449 (revised). Protection from the secondary effects of blast is provided in the usual manner.‡ It is desirable to remove any doors and internal partitions weighing less than 60 pounds per square foot.

The steel "table" or "Morrison" shelter for use in dwellinghouses is a direct application of the principle of ductility.§

For vital factories, the fully framed structure is the only suitable type. Steel and reinforced concrete structures if suitably designed are satisfactorily resistant to collapse.‖

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* For further information on reinforced brick shelters, *see* Home Security Circular No. 290 (1940); also Bulletin No. C.20 of the Research and Experiments Department, Ministry of Home Security.

† For further information, *see* Bulletin No. C.14, Research and Experiments Department, Ministry of Home Security.

‡ For further information, *see* Bulletin No. C.13, Research and Experiments Department, Ministry of Home Security.

§ For further information, *see* Bulletin No. C.16, Research and Experiments Department; and the pamphlet "Shelter at Home"; both issued by the Ministry of Home Security.

‖ For further information, *see* Wartime Building Bulletin No. 17 (H. M. Stationery Office 1941).

Most fully steel-framed single-story factory buildings are reasonably resistant to structural failure. The damage from a near miss or direct hit is usually confined to the lighter elements, the structural framework escaping undamaged. It is important to design so that collapse of structural steelwork will not spread from the failure of a single stanchion.*

A bomb hitting a single-story factory may explode on the roof, between the roof and floor, on the floor, or below it. Except in the last event, fragments fly all around, injuring men and machines. An important function of protective devices is to limit the effects of these bomb fragments and of objects thrown about by blast, by reducing the vulnerable area, that is, the area within which a bomb must fall to cause injury. To protect any comparatively small but very vital item, a portal-framed structure is most suitable. Similarly, reinforced concrete arch structures are used to cover turbo-alternators and other electric plant. Walls, without reinforcement, carrying a heavy slab roof are quite unsuitable, as also are structures incorporating panels of thin light material.

Overhead cover offers protection against bombs exploding in the air almost overhead, and from crater debris. In lofty sheds, portal structures are often the best solution, but they are relatively costly and not suitable for extensive use. In practice, walls are the most useful form of protection. A protective wall should stop bomb fragments and crater debris flung laterally; it should not disintegrate under blast or be overturned by earth shock. These requirements are met by reinforced concrete walls 12 inches thick, or reinforced brick walls 13½ inches thick, with adequate foundation or width of base. For factories operating on line production, a special form of movable wall has been designed. Fire-stop walls should, whenever possible, function also as protective walls and should, therefore, be designed as such. Though a fire-fighting organization remains the chief safeguard from damage by fire, its task is eased by structural barriers opposing fire spread.

The criterion of proper protection is that, having regard to war hazards, the total probable output, over a period, of all existing factories shall be thereby increased by a greater amount than if the work had gone into building new factories which have to be equipped with machine tools. The production methods in the factory must be studied by the planner who must appreciate the degree of importance of each part of the factory and of each item of plant in it. Whatever degree of protection is desired, it must be provided economically; and a simple but sound guide to height and spacing of wall systems has been worked out.

The economic standard of protection depends ultimately on the scale of air attack, but the proportion of material and labor which should be used on protection is independent of the total amount available after provision has been made for housing entirely new processes.

Survey of the Gas Contamination Problem in the Engineering Industry.

The principles of gas decontamination are well established; but in applying them to industrial plant the effects of gas contamination should not be exaggerated.

The survey deals with the effects of gas contamination on the properties of materials and the functioning of machines, and not on the personnel operating the machines, though the danger to the latter resulting from vapor from contaminated machines is considerable.

* For further particulars, *see* Bulletins Nos. C.5, 8, 12, 15, and 19, Research and Experiments Department, Ministry of Home Security.

The chief danger in factories is the splashing of machines and installations with persistent gases, the area of contamination being fairly localized. Roughly 120 square yards are grossly contaminated by a 50 Kg. explosive gas bomb. These bombs have little disruptive effect. Protection against splashing, which can be localized by protective walls, screens, hoods, or even a simple cover for vital machines, is relatively simple.

The chief concerns in the event of gas contamination are: the area and degree of concentration; the maintenance of production; and the danger both to operators working contaminated machines if the running of the plant is continued, and also to operators in adjacent non-contaminated parts of the building.

Much plant can be contaminated, without being otherwise damaged; thus its mechanical functioning need not be prevented. The chief danger is to operators exposed to vapor. Concentration of vapor depends upon the temperature and humidity of the atmosphere, and on air movements. Adequate ventilation is required to deal with concentrations of vapor.

Particular problems arise with the insulation of electrical machinery. As there is no marked corrosion of metal, parts of metal and other non-porous materials can be dealt with by repeated swabbing with solvents. Porous materials may be decontaminated by bleach paste and weathering; the latter is simplest so long as there is no danger of contact with personnel.

An important factor in electrical work is the specific resistivity of liquid gases owing to the danger of flashovers. Experiments suggest that there is little danger of this even when there is a direct path of the contaminating liquid between electrodes. Gross superficial contamination of rotating electrical machinery may be removed by swabbing, residual contaminaton will be removed by running the machine. Even when field coils and armatures are contaminated, the penetration of the insulation will not be considerable if running is continued.

Out-of-door substations and transforming plant are more liable to contamination. Much of the plant can be left to weather after removing the liquid from accessible parts.

Contamination on secondary batteries of the lead-acid type can only occur if a bomb enters the battery room. Spray arresters on the cells provide some protection, which may be increased by the provision of another larger cover above, and clear of, the spray arresters. Interference with the functioning of cells not physically damaged is unlikely, any visible liquid being removed by swabbing with solvents. When the electrolyte is contaminated it will be necessary to renew the acid. Good ventilation is necessary to avoid concentrations of vapor and contamination of neighboring rooms.

The operation of control panels should not be interfered with by splash contamination which may be dealt with by systematic treatment with solvents. Exposed insulated cables and wiring can be treated with bleach paste, while portions of the wiring with absorbent insulating materials must, if grossly contaminated, be cut out, though there should be no immediate breakdown of insulation. Lighter contaminations can be left to weather, provided that attention is paid to the danger from vapor.

Delicate instruments such as relays, voltmeters, and ammeters are usually well protected, and if undamaged the casings can be treated with solvents. When damaged they present a "salvage-cum-decontamination" problem.

Much could be done to localize contamination in open type telephone exchanges by the use of partitions, screens, and curtains. Automatic apparatus will probably continue to function, vapor danger to the staff being the chief factor to guard against. Weathering would be helped by

suitably placed heaters, and some form of extractor could be used with advantage.

The exposed portions of manual switchboards must be treated rapidly with solvents. If the liquid reaches the more vital parts there is little use in trying to decontaminate them. The problem becomes one of salvage.

In factories producing electrical equipment, the multiplicity of stores and partially assembled components require particular attention. Stores generally and individual items in stores should be protected; and it is simpler to remove cartons that become splashed than to decontaminate a large number of components. Small items may be decontaminated by immersion in briskly boiling water for 30 minutes or by hot air, unless they are very heavily contaminated.

It is probably best to concentrate on restoring the assembly shop to working order, and not to attempt detailed decontamination of complicated assembled parts. These should be treated as a salvage problem.

Like blister gases, tear gases are easily absorbed by porous materials. They are generally more corrosive to metals than is mustard gas, and should be removed by swabbing with solvents or by treatment with caustic soda in methylated spirit. Porous materials will have to be left to weather for long periods. Large areas, such as floors, could receive an application of lime slurry.

In conclusion, the chances of contamination causing serious dislocation of industry are not high; and risks can be appreciably reduced by relatively simple means. Where contamination of plant occurs, it need not mean closing the works for long periods. It is advisable to replace the intricate and complicated instruments and components of plant that become contaminated. Emphasis should lie on putting the plant into operation, work in progress, which has been contaminated, being dealt with as a salvage operation.

FITS AND TOLERANCES.—Tables and diagrams prepared by Dr. Ernesto Geiger, show examples of current practice in specification of fits and tolerances. The next article includes a nomographic chart from which may be determined required tolerances and allowances for different classes of fits for various nominal dimensions encountered in engineering practice. Both articles are reprinted from Product Engineering for May, 1942.

Some examples of current practice in specification of fits and tolerances. Both the American tentative standard, referred to as ATS, and the International Standards Association system, known as ISA, are indicated. Although only ATS class numbers are shown on diagram, the ISA designations in the following table were originally specified. Common applications for each example of fit are also tabulated.

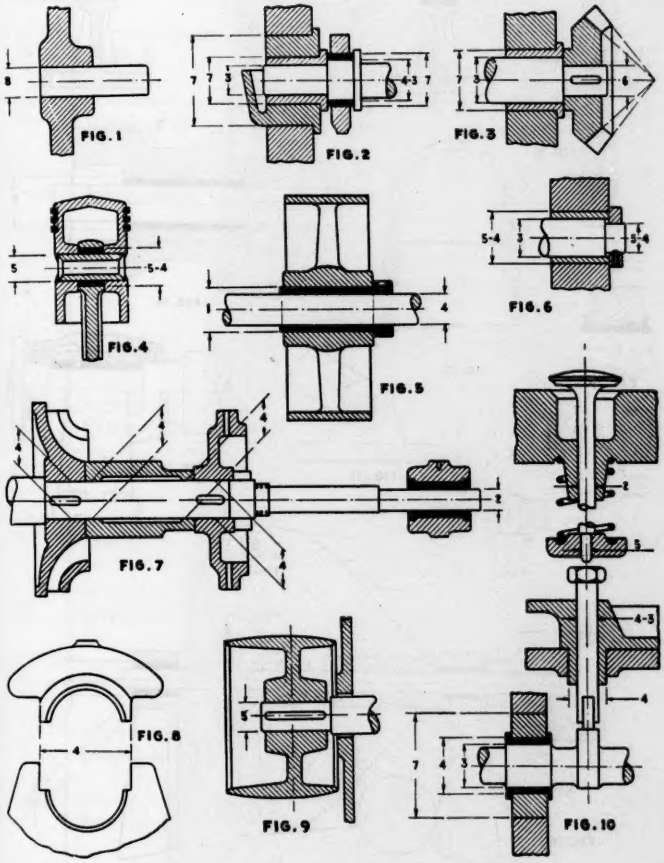
FIG. 1
FIG. 2
FIG. 3
FIG. 4
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FIG. 6
FIG. 7
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FIG. 10

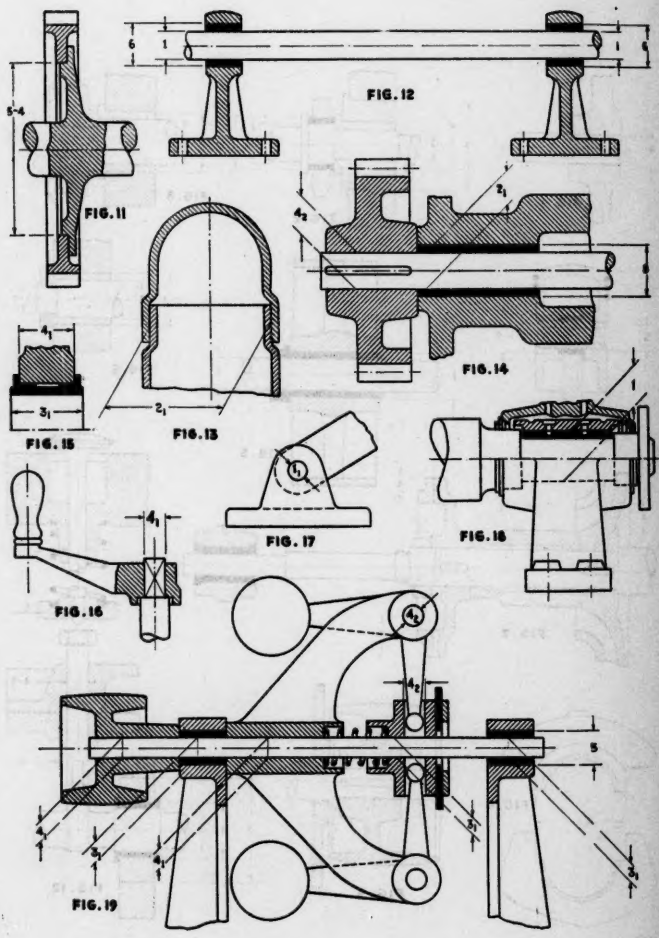
FIG. 11
FIG. 12
FIG. 13
FIG. 14
FIG. 15
FIG. 16
FIG. 17
FIG. 18
FIG. 19

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ATS Class [1]	Name	Desig
8	Heavy force fit	H
7	Medium force fit	H
6	Tight fit	H
5	Wringing fit	H
5–4	Intermediate	H
4	Snug fit	H
4_1	Large tolerance	H8
4_2	Extreme tolerance	H1
4–3	Intermediate	H7
3	Medium fit	H7
3_1	Large tolerance	H9
2	Free fit	H8
2_1	Large tolerance	H10
1	Loose fit	H8
1_1	Large tolerance	H11

[1] See ATS B4a — 1925, "Tolerances
ican Standards Association, 29 W. 3
[2] See "Tolerances for Cylindrical Fit

Classes of Fits, Designations and Applications

ISA Designation [3]	Applications of Fits Illustrated
H7r6	Wrist pins, armature shafts in electric motors, shaft couplings and flanges (Figs. 1, 14).
H7n6	Bushings, friction clutch disks, engine cranks, cylinder liners, eccentrics on their axles, and wheels of locomotives and cars (Figs. 2, 3).
H7m6	Keyed gear hubs, axially stressed bushings, jig bushings, pulleys and rocker arms (Figs. 3, 12).
H7k6	Keyed hubs of gears and pulleys, keyed shaft couplings, handwheels and cranks, locating keys, tongues and dowels (Figs. 4, 9, 10, 19).
H7j6	Centrifugal rotors, pulley hubs, centering pins and rims, and renewable bushings and collars (Figs. 4, 6, 11).
H7h6	Automotive shift gear hubs, milling cutters on mandrels, centering pins and rims for frequently dismantled machines, centrifugal pump rotors, interchangeable bushings, bearing caps, shims and body and jaws of drill chuck (Figs. 5, 7, 8, 10).
H8h8	Agricultural and mining machinery, interchangeable parts not rotating on shafts, stop collars for shafts, hand cranks and heavy shaft couplings (Figs. 15, 16, 19).
H11h11	Heavy machinery such as hoists, cranes, etc. (Figs. 14, 19).
H7g6	Cross heads in their slides, sliding gears, valve plungers and valve lifters, and eccentrics in their collars. (Figs. 2, 10).
H7f7	Shafts rotating at less than 600 r.p.m. with bearing loads under 600 lb. per sq. in. (Figs. 2, 3, 6, 10).
H9f8	Bushings for vehicle wheels, heavy bearings, hydraulic pistons and bearing width (Figs. 15, 19).
H8e7	Bushings on multiple-supported transmission line shafts, crankshaft bearings, oscillating bearings and quick acting slides (Figs. 7, 10).
H10e9	Material handling machinery and stamped pieces from metal sheets. Used also on electric switches (Figs. 13, 14).
H8d10	Idle pulley bushing quick acting plungers, valve lifters and turbo-generator axle-bearings (Figs. 5, 12, 18).
H11d11	Hoists and other material handling machinery (Fig. 17).

, "Tolerances, Allowances and Gages for Metal Fits," American Society of Mechanical Engineers and ation, 29 W. 39th St., New York; price 50 cents.
Cylindrical Fits" by John Gaillard *Industrial Standardization*, January to April, 1941.

TOLERANCES AND ALLOWANCES.—This companion piece to the preceding article is also taken from Product Engineering, May, 1942.

Fits and tolerances of holes and shafts or other mating parts, as computed from American tentative standard B4a—1925 of the American Standards Association, can be determined quickly with the chart on a following page and a straight edge. Straight lines through the selected value on the nominal dimension scale at the left and two points in the center of the chart will indicate desired allowances and tolerances on the right-hand scale.

The 11 points in the center of the chart are numbered to correspond with the classes of fit for which they are used and are labeled to indicate the result or results to be obtained from each on the right-hand scale. Allowances for classes 4 and 5 are zero for all nominal dimensions of mating parts and, therefore, are not found on the chart. Only two solution lines or two positions of a straight edge are required for any one nominal dimension.

The right-hand scale is read only to the closest ten-thousandth, as indicated by the following examples which may be checked on the chart on the following page. Dimensions are given in the standard basic hole system.

Recommended Applications of Classes for Interchangeable Manufacture.

Class 1: Loose fit, large allowance. Provides considerable freedom: used where accuracy is not essential. Employed on agricultural and mining machinery; controlling apparatus for marine work: textile, rubber, candy, and bread machinery; and general machinery of a similar grade.

Class 2: Free fit, liberal allowance. For running fits where speeds of 600 Rpm. or over and general pressure of 600 pounds per square inch and over. Intended for use in electrical machinery, engines, many machine tool parts, and some automotive parts.

Class 3: Medium fit, medium allowance. For running fits under 600 Rpm. and with general pressures less than 600 pounds per square inch; also for sliding fits, the more accurate machine tools, and automobile parts.

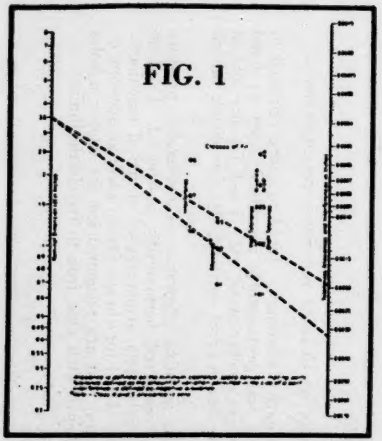
Class 4: Snug fits, zero allowance. Closest fit that can be assembled by hand and necessitates work of considerable precision. Used where no perceptible shake is permissible and where moving parts are not intended to move freely under load.

Class 5: Wringing fits, zero to negative allowance. Assembly is usually selective and not interchangeable.

Recommended Applications of Classes for Selective Assembly.

Class 6: Tight fit, slight negative allowance. Light pressure required to assemble. Parts are more or less permanently assembled, such as fixed ends of studs, hook eyes, pulleys, rocker-arms, etc. Used for dry fits in thin sections or extremely long fits in other sections, also for shrink fits on very light sections.

Class 7: Medium force fit, negative allowance. Considerable pressure required to assemble, parts considered permanently assembled. Used in fastening locomotive wheels, car wheels, armatures of electrical machinery, and crank disks to their axles. Used for shrink fits on medium sections or long fits. These fits are tightest that are recommended for cast-iron holes or external members since they stretch cast iron to its elastic limit.

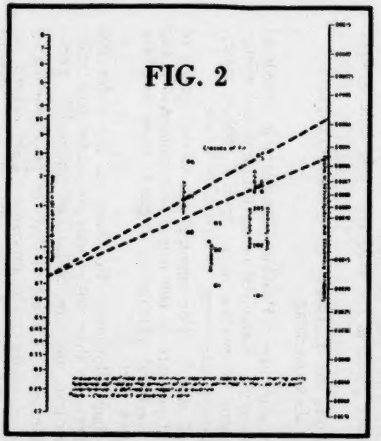


Example 1: 3½-in. shaft and hole, Class 2 fit. (see Fig. 1)

Shaft and hole tolerance = 0.0020 in.

Allowance = 0.0033 in.

	Dimensions		
Hole	$3.5000 \begin{smallmatrix}+0.0020\\-0.0000\end{smallmatrix}$	or	$\begin{smallmatrix}3.5000\\3.5020\end{smallmatrix}$
Shaft	$3.5000 - 0.0033$		
	$= 3.4967 \begin{smallmatrix}+0.0000\\-0.0020\end{smallmatrix}$	or	$\begin{smallmatrix}3.4967\\3.4947\end{smallmatrix}$

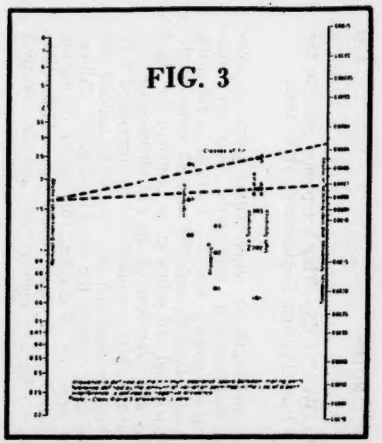


Example 2: ¾-in. shaft and hole, Class 7 fit. (see Fig. 2)

Shaft and hole tolerance = 0.0005 in.

Interference = 0.0004 in.

	Dimensions		
Hole	$0.7500 \begin{smallmatrix}+0.0005\\-0.0000\end{smallmatrix}$	or	$\begin{smallmatrix}0.7500\\0.7505\end{smallmatrix}$
Shaft	$0.7504 \begin{smallmatrix}+0.0005\\-0.0000\end{smallmatrix}$	or	$\begin{smallmatrix}0.7504\\0.7509\end{smallmatrix}$



Example 3: 1.625-in. shaft and hole, Class 4 fit. (see Fig. 3)

Shaft tolerance = 0.0005 in.

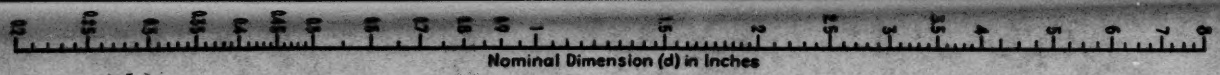
Hole tolerance = 0.0007 in.

Allowance = 0.000

	Dimensions		
Hole	$1.6250 \begin{smallmatrix}+0.0007\\-0.0000\end{smallmatrix}$	or	$\begin{smallmatrix}1.6250\\1.6257\end{smallmatrix}$
Shaft	$6.6250 \begin{smallmatrix}+0.0000\\-0.0005\end{smallmatrix}$	or	$\begin{smallmatrix}1.6245\\1.6250\end{smallmatrix}$

−0.0020 or 3.4947

Shaft 0.6250 −0.0005 or 1.6250



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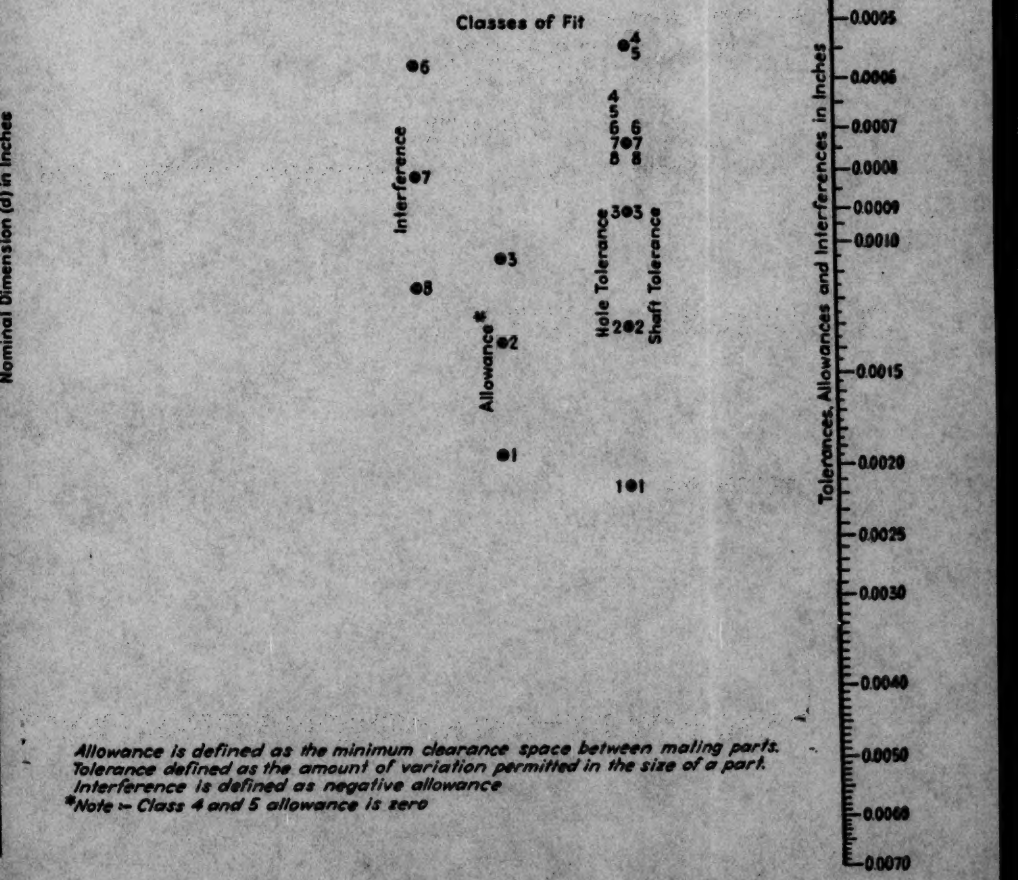
TOLERANCES AND ALLOWANCES

Nomographic Determination

American Tentative Standard A.S.A. B4a — 1925



Allowance is defined as the minimum clearance space between mating parts.
Tolerance defined as the amount of variation permitted in the size of a part.
Interference is defined as negative allowance
**Note — Class 4 and 5 allowance is zero*

Class 8: Heavy force and shrink fits, considerable negative allowance. Used for steel holes where metal can be highly stressed without exceeding elastic limit.

SPARKS FROM ALUMINUM-PAINTED STEEL.—An abstract of T. S. E. Thomas' article "Ignition by Sparks from Aluminum-Coated Metalwork," published by Foundry Trade Journal, September 25, 1941, is reprinted from Metals and Alloys, January, 1942.

Attention is called to the danger of heat treatment after painting. It has been reported that sparks have been observed in certain industrial establishments when steam pipes and steam-heated vessels coated with aluminum paint have been struck or scraped with a tool of steel, copper, bronze or brass. Only a light blow or scrape was found to be necessary to produce the effect.

Experimental work on the subject has brought out several points. First, the effect occurs only when the steel surface before being painted is rusty; careful preliminary cleaning eliminates the sparking. After the rusty specimen has been painted, it must be subjected to a temperature in the neighborhood of 265 degrees F., before sparking can be obtained.

No sparks could be obtained from a specimen of rusty steel, painted with an ordinary oil-vehicle aluminum paint, which had not been heated though the specimen had been kept for 1 year at laboratory temperature before being tested. Once the specimen has been heated after being painted, the temperature of the steel when struck has little or no effect on the character of the sparks obtained, *i.e.*, the effect achieved by suitable heat-treatment persists at normal temperature.

It was suggested that the phenomenon was a miniature thermit reaction between flakes of aluminum and the particles of rust, which are intermingled at the interface, the reaction being started by the heat generated at the point of impact between the metal tool and the iron surface. Also, with painted metal that had not been heated before trial, the vehicle in the paint acted as a lubricant—so reducing the temperature rise at the point of impact and at the same time preventing intimate contact between aluminum flakes and particles of rust, that the effect on the specimen of initial heating was the destruction of the vehicle and the consequent elimination of its preventive role.

Tests made with ordinary oil-vehicle aluminum paint, with paints containing a cellulose base, etc. are discussed. The use of aluminum paint to cover rusty steelwork does not give rise to any danger from sparks, on impact with a piece of hard metal, if commercially-produced paints of the types described are used, unless the metal is heated to 300 degrees F., or upwards subsequent to being painted. However some incorrectly made paints can be dangerous without heating of the metalwork.

BOOK REVIEW.

MARINE DIESEL HANDBOOK, By Louis R. Ford. Published by Diesel Publications, Inc., 192 Lexington Ave., New York City. $7.00 in the United States; $8.00 Elsewhere. 823 Pages.

The author, Louis R. Ford, M.A., is well known for his many writings on the subject of Diesel engines. He is Editor of Motorship and Diesel Boating, former Associate in Mechanical Engineering, Columbia University, Ex-Lieutenant Commander, U. S. Navy, and a member of this Society among his numerous achievements.

The volume is beautifully printed and generously illustrated. In the words of the author it "has been prepared as a compendium of information regarding the design, construction and operation of the marine Diesel engine, in connection with the miscellaneous other items that constitute the ordinary equipment of a workable motor vessel."

The 27 chapters, beginning with "Marine Applications of the Diesel Engine" and "Principles of the Diesel Engine" carry through all phases of the subject to a final chapter containing "Practical Questions on Operation."

OTHER BOOKS RECEIVED

These books, although read with interest, are not reviewed because their nature is non-engineering.

VICTORY IN THE PACIFIC, By Alexander Kiralfy, Published by the John Day Company, New York, 282 Pages.

THE TRAGIC CAREER OF COMMODORE JAMES BARRON, By Paul Barron Watson. Published by Coward-McCann, Inc., New York, 84 Pages.

ASSOCIATION NOTES.

MEMBERSHIP.

The following have joined the Society since the publication of the May 1942, JOURNAL:

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